OXC-0414-60
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THERMAL SHUTTER

FEASIBILITY AND DESIGN STUDY

P.A.E.

March 8, 1960

Document 135

STATINTL

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pages 69 + ii

SUMMARY

The detailed design for a shutter mechanism which is presented appears practical. The thermal effectiveness of the shutter is shown to be about equivalent to two low emissivity coatings.

It is shown that the use of a thermal shutter involves a relatively large weight penalty. Although required mounting size increases, the scan reduction penalty is negligible. It is felt that mechanization of the shutter in vacuum is achievable.

TABLE OF CONTENTS

I. INTRODUCTION

The scanning nature of the "T System" lends itself to the concept of occluding the photographic window except during the scan cycle. If the occluding mechanism presents a highly reflective surface to incoming thermal radiation, incorporation of such a mechanism reduces the total heat flux passing through the window into the equipment bay.

The feasibility of such a mechanism was investigated. The implications of utilizing such a thermal shutter were analyzed. The design of a suitable shutter mechanism was undertaken and design integration with the window system was accomplished. This report serves as a summary of these activities.

II. IMPLICATIONS OF USING A THERMAL SHUTTER

This section contains a discussion and analysis of the most salient aspects which must be considered in evaluating the incorporation of a thermal shutter as part of the window configuration.

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Weight:

Design studies place the weight of each shutter mechanism at about 12 pounds. Two separate shutters would be required, one for each window system. In addition, there will be a small increase in weight of the window mount.

Size:

Because the thermal shutter requires a cooling bar and rollers to be located at the sides of the shutter gap, the overall width of the window system must be increased over the width required for a shutterless system. For the present hatch contours the maximum width of the window system can be 22.75 inches. Designs indicated that 6.5 inches in addition to the free aperture of the window are required for mounting a window incorporating a shutter. Figure II-1 shows the resultant free aperture of the outer glazing and corresponding unvignetted angle of scan for the "T System" located with the scanning prism at various distances from the inner glazing of a three glazing window system. Present design parameters indicate that the prism may be located 1.25 inches from the inner glazing. This results in a scan reduction of about 1 degree on each side.

Reliability:

It is felt that the reliability of the thermal shutter mechanism can be made consistent with the reliability of the overall system. However, since it is an active system, a window with a shutter is invariably less reliable than a purely passive, shutterless window system.

WINDOW CLEAR APERATURE DIMENSIONS AS A FUNCTION OF PRISM TO WINDOW CLEARANCE FOR A THREE GLAZING CONFIGURATION.

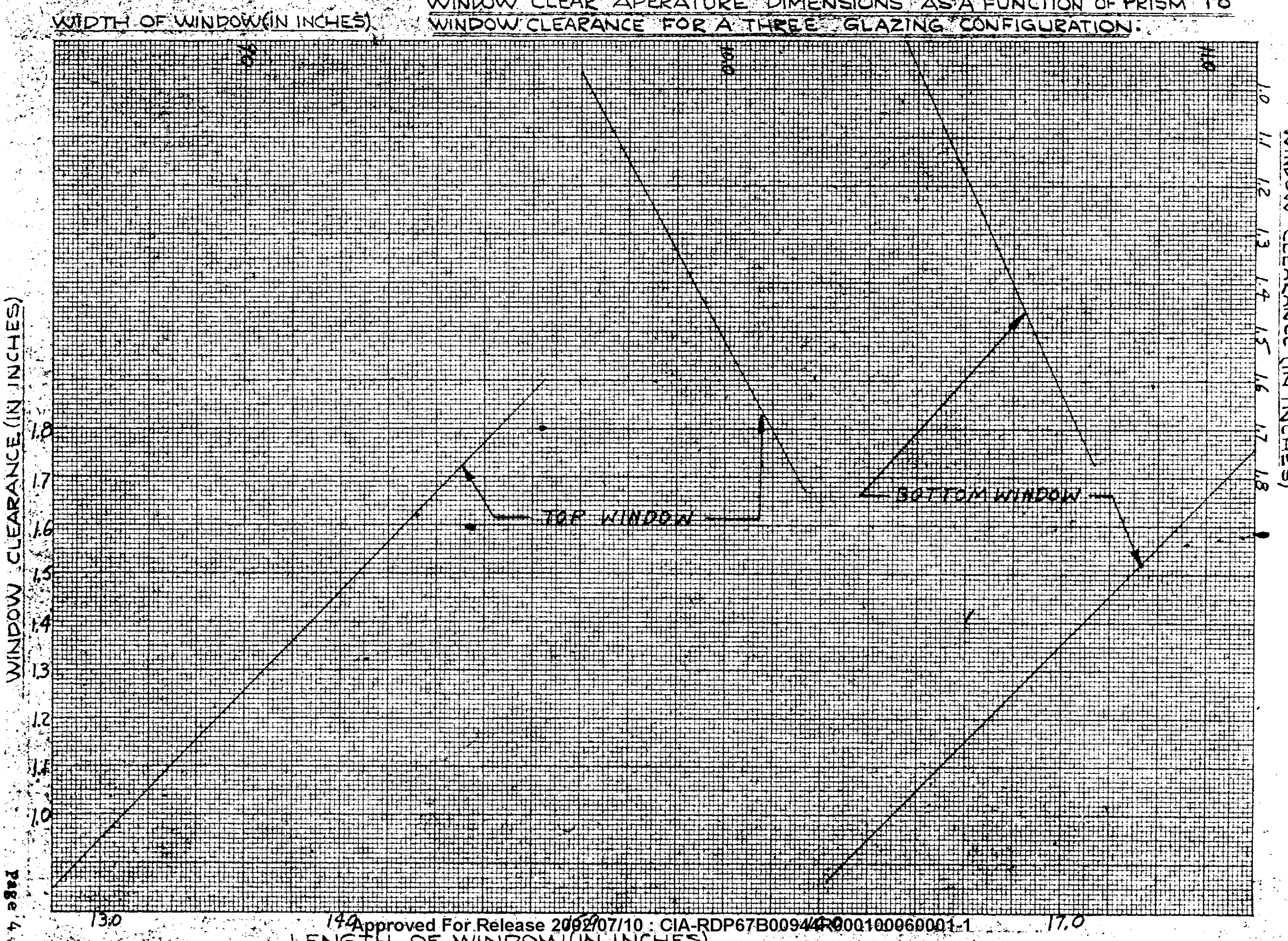

FIGURE II-1

Location of Shutter Mechanism:

The function of the shutter is to intercept radient energy and minimize the amount of heat flux entering the equipment bay. In determining the location of a thermal shutter, the configuration of the window system must be examined. In a three glazing configuration, the cooling air located in the inner gap is used to remove the heat flux which penetrates the two outer glazings. Since the third glazing is itself relatively opaque of the wavelengths of the incident heat radiation, the location of the reflective shutter in this cooling would serve no useful purpose. The outer vacuum gap is designed to eliminate heat transfer by conduction and convection. The shutter installed in this gap would serve to partially limit heat transfer by radiation across the gap. This would result in a lower temperature of the middle glazing, and would reduce the cooling air requirement in the inner gap. The relative effectiveness of a thermal shutter located in the vacuum gap is analyzed in a following section.

Mechanism in Vacuum:

The necessary location of the shutter in the vacuum gap requires special attention to two aspects. The drive mechanism must pass through the vacuum seal without impairing the quality of the vacuum. Furthermore, bearings capable of operating in the hot-vacuum environment must be used. Preliminary studies indicate that the use of a hermoflex as a pass-through mechanism is satisfactory. The technology of bearing lubrication in a vacuum has recently been advanced to the point where satisfactory lubricants are now available. However, the primary effort by bearing manufacturers has not concerned itself with the out-gassing effects of these lubricants, since the intended use has been for the infinite vacuum of outer space. Barden has been consulted and has provided samples of bearings which they feel will not out-gas enough to destroy the vacuum. These bearings are going to be tested in our laboratory.

Mounting Considerations:

The installation of the shutter in the vacuum gap makes the mounting of the glazings a more complex problem, since the ring which normally separates the glazings at the edges must be eliminated.

Vacuum Seal:

The sealing of the two glazings to accomplish a permanent hard vacuum has not yet been accomplished for either the shutter or non-shutter window system. The use of an ion exchange or gettering pump may be required to maintain the desired vacuum level in either system. If such an auxilliary pump is required, introduction of the shutter mechanism should not increase the complexity of the vacuum problem.

Thermal Effect of Scanning Aperture:

The glazing interior to the shutter is heated as the shutter aperture scans across it. This has the effect of producing a thermal wave which can be considered to travel along the length of the glazing. The exposure of each image point is accomplished by rays traveling through this thermal wave. However, since the exposure time is in the order of 1/50 second, the movement of the thermal wave may be considered negligible as seen by each image point. Exactly what this optical effect will be has not been calculated.

The temperature rise of the glazing due to each shutter scan has been calculated as a function of thickness. For a configuration containing one low emissivity coating the temperature rise at the trailing edge

of the shutter aperture varies linearly from about 4.5 to 0.9 farenheit degrees from 0.01 to .5 inches thickness respectively.

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III ADVANTAGES OF INCORPORATING A THERMAL SHUTTER

This section presents the salient advantages which are afforded by incorporating a thermal shutter into the window configuration.

The incorporation of a thermal shutter affords two principle advantages. By its use, a reduction in the required number of low emissivity coatings can be affected without increasing the total heat transfer through the window system. On the other hand, incorporation of the shutter without elimination of the coatings, reduces the heat transfer to the inner glazing.

The effectiveness of the shutter when used with various coatings is compared to the shutterless case in the table below, where the quantities given are the resulting effective emissivity of each combination. A complete analysis is contained in Appendix B.

Surface Emissivity / Condition	ϵ_1 .2	ϵ_2 .2	ϵ_1 .2	ϵ_2 .8	ϵ_1 .8	ϵ_2 .8
No Shutter	.11		.19		.67	
Shutter	.019		.032		.11	

In the case where heat transfer to the inner glazing is reduced, the heat removal requirement upon the cooling air in a three window configuration is reduced. This can manifest itself by the proper manipulation of the cooling air parameters, in a reduced lateral thermal gradient.

In the two window case, when no cooling air gap is provided, the reduction of heat transfer resulting from incorporating a shutter, will maintain the interior glazing at a lower steady state temperature than if no shutter were present.

IV SPECIFICATIONS USED IN DESIGN OF THERMAL SHUTTER

This section contains the specifications for the shutter design which were released to subcontrastors. These specifications were supplimented by verbal communications and frequent design meetings. This approach was used in order to maintain shutter design consistent with the changing mounting and space requirements.

Approved For Release 2002/07/10 : CIA-RDP67B00944R000100060001-1

GENERAL SPECIFICATIONS FOR THE

RECIPROCATING CURTAIN MECHANISM

Prepared by: [redacted] STATINTL

Date: January 18, 1960

GENERAL SPECIFICATIONS FOR THE RECIPROCATING CURTAIN MECHANISM

Environment

The curtain is located in a vacuum area. Associated equipment must necessarily also be located in an extension of this vacuum area. The degree of vacuum is of the order of 10^{-3} to 10^{-5} mm mercury.

The mechanical mount or frame which contains the curtain mechanism will be subject to a severe thermal environment. The anticipated thermal range is -20° to 300° F. The curtain will operate at the upper end of this range during the major portion of its use. If possible, 500° F instead of 300° F should be considered as the operating environment.

Curtain Function

The curtain will be required to pass between two pieces of glass. At no time can the curtain be allowed to contact either piece of glass.

An aperture in the curtain is required when the curtain is travelling in one direction. When the curtain is returned, the aperture should be closed. The size of the required aperture should vary for different positions across the glass. This variation in aperture will be a function which can be built into the mechanism and will always be the same as a function of position along the glass.

The speed of the curtain or rate at which the aperture must traverse the glass must be made continuously variable. The synchronism and rate will be controlled by an external input, which is discussed in more detail under the "Curtain Position Synchronization" section. The approximate range of curtain rates are 2.7 to 8.3 inches per second.

Curtain and Aperture Program

A plot of position with respect to one edge of the glass plate, for each edge of the aperture as a function of principal shaft rotation, is the manner in which program data shall be presented for design of the controlling mechanism. Such a plot will allow determination of non-linearity of either edge of aperture. with respect to position along the glass plate and will allow calculation of required aperture size with respect to position along the glass plate.

Direction of Travel

Present requirements are such that two models of the curtain mechanisms may be required. One model shall be required to operate with the aperture travelling from the centerline (Figure 2) outward. The other will be required to operate in the opposite direction; from outward toward the centerline. Consideration should be given to the possibility of incorporating this requirement into one basic mechanism which can be set to function one way or the other. A possible solution would be to make the shutter package symmetrical so that it would fit into the mounting frame with either end toward the centerline.

Time of Cycle

A complete cycle is defined as a traverse of the curtain across the glass plate and the subsequent return to the starting position. The required traverse of the open aperture begins with the trailing edge of the curtain aperture coincident with one edge of the clear aperture of the glass plate and ends with the leading edge of the curtain aperture coincident with the other edge of the clear aperture of the glass plate. This part of the complete cycle can be referred to as the exposure period. The remaining portion of the cycle can be referred to as the rewind period. The relation of the exposure period and rewind period to the complete cycle is given by:

$$\text{Exposure Period} = \frac{88}{180} \text{ cycle}$$

$$\text{Rewind Period} = \frac{92}{180} \text{ cycle}$$

Cycle time varies approximately from 3.6 to 11.2 seconds.

Modes of Operation

The curtain mechanism will be required to operate in two modes of operation. Mode I will be operation of the curtain between the two glass plates, but the aperture will remain closed throughout travel in both directions. Mode II will be operation of the curtain with the aperture functioning according to the programmed pattern during travel in one direction only. It shall remain closed during the return sweep.

Size of Curtain and Aperture

The size of the glass plate presented in Figures 1 and 2 are only approximate and should be used as a general guide only.

The aperture size will be about 10 inches wide and vary from about 6.5 to 4.5 inches along the direction of travel.

Curtain Position Synchronization

It is recommended that the curtain servo system incorporate synchro resolver for position control. The input signal would be one to one with the principal shaft of the overall system. The curtain is required to perform one cycle per 90 degrees rotation of the principal shaft. Hence, the curtain should perform four complete cycles per one resolver revolution. Since the input would be linear, the curtain mechanism would be required to generate the non-linear control function.

For purposes of prototype manufacture, the following should be considered:

Kearfott Co., Inc. - Synchro resolver #R 982-012, specified for 200° C operation.

Accessibility

When installed for operation, the mechanism will be entirely inaccessible for adjustment, observation or control. All control functions must be made automatic and adjustments preset, except for the following external controls and adjustments:

Off-On

Mode I - Mode II

Curtain position synchronization signal

These functions will be controlled by incorporation with the overall system.

Cooling of Curtain

The curtain itself will intercept infrared radiation and will, therefore, be subject to heating. Provision should be made to be able to remove this heat, in order to maintain the curtains at constant temperature, both with respect to its length and with respect to time. Cool air will be available to pass through sealed rollers or shoes.

Coating of Curtain

The curtain must be made of such a material and be so designed as to allow a thin coating to be applied to the lower surface. The purpose of such a coating will be to act as an infrared reflector. An evaporated aluminum coating may be used. Precuations should be taken so that such a coating will not be scratched or rubbed off.

If necessary, the upper side of the curtains may be coated with a material of high thermal conductivity. The purpose of such a coating would be to aid in removing any accumulated heat from the curtain to a sink.

Size and Weight

Figure 2 will serve as a preliminary guide to available space for the curtain mechanism. As the mounting design progresses, coordination with respect to space allocation will have to be maintained.

A design objective should be minimimum weight.

Additional Space

Additional space is available outside the vacuum area, but remote from the curtain mechanism, for placement of electronic devices, if required. The environment in such an area is about 5 psia and 120° F.

Vibration

The entire curtain mechanism may be subject to a severe vibrational environment during its operation. More data on the frequency and amplitude of this vibration will be submitted if and when required.

An attempt should be made to maintain curtain mechanism operation as vibration free as possible, and strong transient shocks should be avoided.

Power

28 volts d.c. is available as a power source. A limited amount of 400 cycle, 115 volt power may also be available, but should not be considered for this application unless of significant advantage.

Reliability and Life

The curtain must be designed in such a manner that there can never be a failure during actual operation. A failure is defined as a malfunction which would either cause the curtain to break or remain open, or which would cause the aperture to lose position synchronization with the input signal. Loss of vacuum also constitutes failure.

Preoperational servicing to insure such reliability is possible. The minimum number of cycles between servicings must be at least 4000. Life of the mechanism should be at least 100,000 cycles.

Additional Information

The following additional data will be made available as soon as possible.

1. Plot of leading and trailing edge of aperture vs. principal shaft rotation.
2. Size of glass plates and required clear aperture.
3. Firm date on required aperture velocity range.
4. Firm date on required vacuum.
5. Additional drawings of space availability based on coordinated space requirements.
6. Firm data on cycle time.

ISOMETRIC VIEW OF GLASS ARRANGEMENT

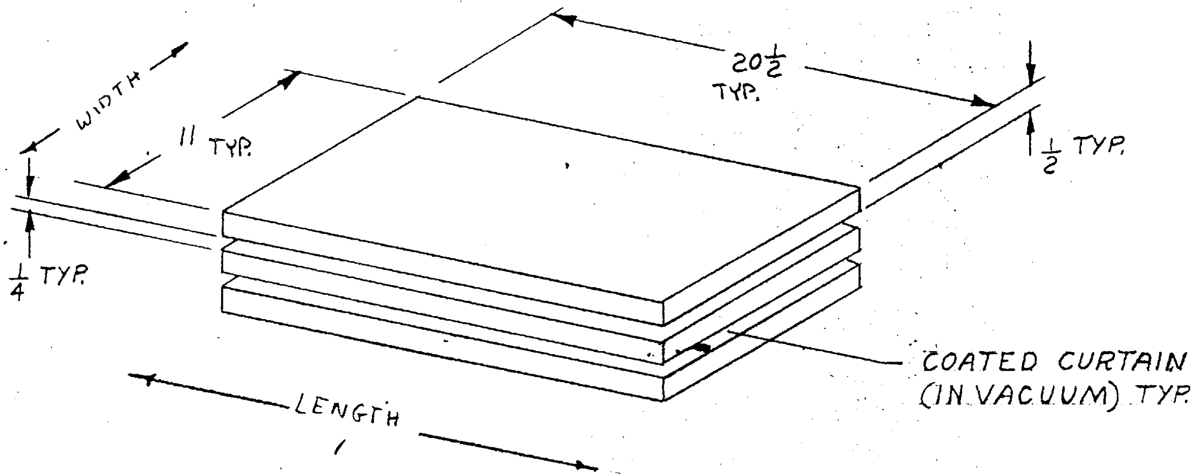


FIGURE 1

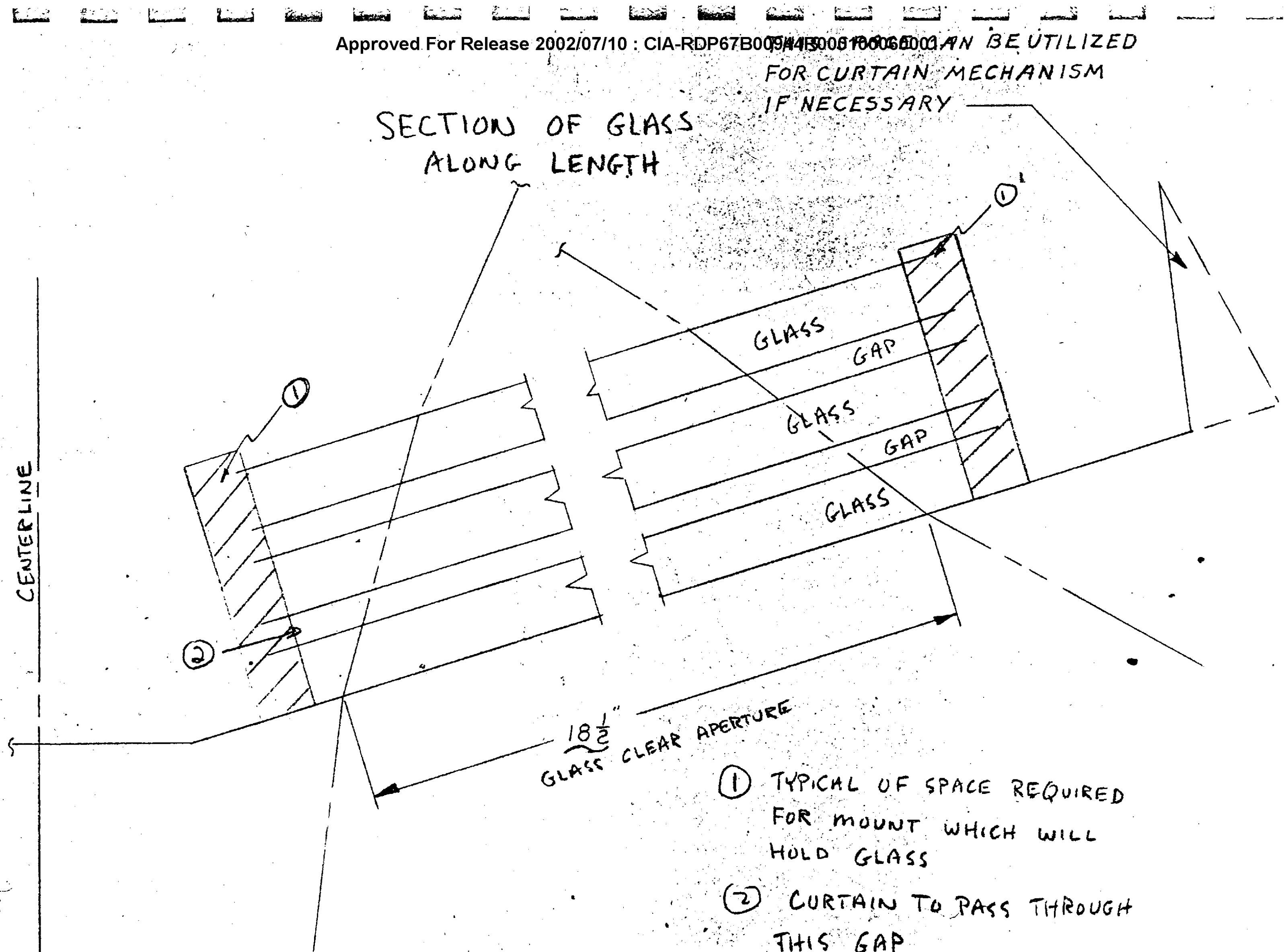


FIGURE 2

Page 19

V PROPOSED DESIGN OF THERMAL SHUTTER

By: [redacted] STATINTL

This section contains the technical report submitted by

STATINTL [redacted] in fulfillment of the shutter design

subcontract.

[redacted]

[redacted]

February 26, 1960

REPORT ON CURTAIN SHUTTER DESIGN

I. General Description

This shutter is to consist of two curtains of hard rolled stainless steel, .0013 inch thick. Each curtain has a rectangular portion and two ribbons extending from it, as shown in Fig. I. The edge at which the two ribbons join the rectangular portion is reenforced by a slightly thicker metal strip formed as shown in Fig. II.

These curtains will be supported on four rollers in the manner shown in Fig. III. The exposure aperture is formed between the two reenforced edges. By adjusting the two rollers carrying one of the curtains relative to the two other rollers, the aperture can be opened and closed and made to vary in width in a predetermined manner. This adjustment is imparted by the driving mechanism.

The curtain rollers contain long helical springs and are similar in construction to the familiar household shade rollers. This construction is shown in longitudinal section in Fig. IV. These rollers will be supported to rotate on precision ball bearings of a type very recently developed by the Barden Company for use in vacuum.

The curtains will operate in the ¼ inch space between two of the glass plates of the main window assembly. They will be guided at their edges to keep them in the middle of the space even under fairly high accelerational forces.

The springs in the curtain rollers will be wound in opposition to each other so as to place the curtains under a tension of the order of six pounds. This is to be done in order to keep the curtains flat even when they are subjected to vibration and accelerational forces. One of the springs in each pair of rollers will be wound to a tension somewhat higher than the other, so that each curtain tends to move in the direction of the roller containing the more strongly wound spring. The two curtains will tend to move in the same direction.

The driving and controlling mechanism will be attached to the rollers containing the less strongly wound spring. During the exposure part of the cycle this mechanism will operate to move the curtains in the direction opposite to that in which the springs tend to make them move. During the return part of the cycle it will permit the curtains to move back under the influence of the more strongly wound springs. During this return part of the cycle the aperture between the reenforced edges of the curtains will be closed, with the edges overlapping about .25 inch.

The curtains and their supporting rollers, including the helical springs, will be in the evacuated space. The connection between the roller shafts and the driving mechanism will be made through two "Hermeflex" hermetic rotary seals as made by the Kearfott Company, Inc.

The driving and controlling mechanism operates through two cams and cam followers which communicate motion through trains of gears to the Hermeflex couplings and through them to the shutter rollers. These two cams are on a common shaft which is driven one revolution per cycle by a servo motor. A synchro resolver geared to this cam shaft sends a positional signal back to the servo system and thus makes it possible to control the cycling rate of the shutter as desired.

One of the two cams, which may be called the main drive cam, imparts to both curtains the principal forward and backward motion by which the edges of the curtains traverse the entire length of the supporting frame in each cycle. This motion is imparted by direct gearing to one of the curtains, hereafter referred to as the follower curtain. It is imparted to the other curtain, the leading curtain, through a planetary gear differential. The gearing is such that as long as the adjustment of the differential is unchanged the two curtains are caused to travel back and forth with a constant distance between their edges.

The second cam, which may be called the shutter opening cam, operates through the differential to control the opening between the edges of the curtains. During the return movement of the curtains the aperture is closed with the edges slightly overlapped, as stated above. Just before the beginning of the exposure the shutter opening cam rotates the sun gear of the differential in such a direction as to move the leading curtain ahead of the follower curtain a distance equal to the desired initial opening of the shutter. During the exposure this cam operates the sun gear of the differential through small angles sufficient to cause the opening between the two curtains to follow the law expressed in Perkin-Elmer's drawing entitled "Shutter Opening", dated 1/19/60. After the end of the exposure the cam rotates the sun gear in the proper direction to cause the leading curtain to slow down and reverse its motion while the follower curtain continues to move ahead rapidly, thus closing the aperture. During the greater part of the return motion of the curtains the shutter opening cam holds the sun gear of the differential stationary in a position which establishes the desired small overlap of the curtains, shutting out light entirely during this part of the cycle.

The proper time relationship between the two cams is established permanently by the fact that they are attached to the same main drive gear.

An additional control is provided by which the curtain opening cam may be temporarily rendered inactive. This consists of a small auxiliary motor, a gear train, a spring, and a lever urged by the spring which moves the pivot point of the cam follower associated with the curtain opening cam. When this pivot point is in the position to which it is moved by the action of the spring and lever, the cam follower roller is out of contact with the cam during its entire rotation, and the gear to which the cam follower is linked is rotated into contact with a stop which holds it in a position such that the sun gear of the differential is in its position corresponding to the completely closed curtain aperture. This condition is established whenever power is removed from the auxiliary motor. In this state of affairs the main drive cam can cause the two curtains to move back and forth together, but the edges of the curtains will remain overlapped during the entire cycle, that is, no exposure will take place. When power is applied to the auxiliary motor it overpowers the spring and returns the pivot point of the cam follower to the position in which the follower is in contact with the cam and transmits the curtain opening action as previously described.

II. Detailed Description of Drive Mechanism

The drive mechanism as a whole is shown in Fig. VI. Fig. VI contains three views of the drive mechanism. The bottom view shows the motor drive gear train and the main train of gears which actuates the backward and forward movement of the curtains under the control of the main drive cam. The servo motor is mounted at the extreme left. Its shaft carries a 15 tooth, 96 pitch, 20 degree pressure angle gear. A 90 tooth, 96 pitch gear is provided to mesh with this motor pinion, and on the shaft of [illegible] gear an [illegible] tooth, 64 pitch, 20 degree pressure [illegible] gear is mounted. This meshes with a 108 tooth gear which in turn has a 21 tooth, 64 pitch gear mounted on its shaft. This shaft has an auxiliary support plate to carry the bearing in which it is supported on the end nearest to the 21 tooth gear. This support plate is shown in section in the top view on this sheet.

A 112 tooth, 64 diametral pitch gear acts as a bridge between the 21 tooth gear and the 140 tooth, 64 pitch gear which carries the two cams.

The driving ratio between the motor shaft and the 140 tooth gear which carries the two cams is

$$\frac{15}{90} \times \frac{18}{108} \times \frac{21}{140} \text{ which equals } \frac{1}{240}$$

If the servo motor is operated at 4000 rpm, which is at its most efficient point, it will impart to the 140 tooth gear a rotational speed of 16-2/3 rpm, or one revolution in 3.6 seconds. This is the shortest cycle time specified and represents the condition under which maximum power will be required from the servo motor. At slower rates when the motor is operating less efficiently, less power will be required by the mechanism.

The main drive cam which has what might be described as a lop-sided kidney shape drives a sector of a 175 tooth, 64 pitch gear. This sector carries the ½ inch diameter cam follower roller between two side plates which are riveted to the gear. This sector drives a 50 tooth gear which is on the same shaft with a 117 tooth gear. Three further step-ups of speed in the ratio of

117 to 50 bring the motion to a 42 tooth gear whose shaft is linked through one of the Hermeflex couplings to the controlling roller of the following curtain system.

The final 117 tooth gear in this system carries end bearings which support four planetary gears each having 28 teeth, of 64 diametral pitch. A plate slightly smaller in diameter and spaced ¼ inch away from this gear carries the other bearings for these planetary gears. The two plates are spaced apart by four small studs placed between the planetary gears. These studs are riveted into the 117 tooth gear and are threaded at their opposite ends to receive screws which hold the opposing bearing plate in position.

The 117 tooth gear and the 50 tooth gear which is integral with it on one side and the bearing plate on the other side of this assembly carry ball bearings by means of which the assembly can rotate freely on a shaft which carries a 28 tooth, 64 pitch gear which is the sun gear meshing with the four planetaries.

The raised ring on the 117 tooth gear and a similar raised ring on the opposed bearing plate have a space between them of .130 inch in which is placed a ring gear having 117 teeth on its outside diameter and an 84 tooth internal gear on the inside diameter. This 84 tooth gear meshes with the four 28 tooth planetary gears so that this ring gear is carried around by them if the bottom 117 tooth gear rotates and thus transports the planetaries. The ring gear will also revolve around the four planetary gears if the sun gear is rotated. If the sun gear is held fast and the bottom 117 tooth gear is rotated carrying the planetaries around with it, the ring gear will make four revolutions for every three revolutions made by the bottom 117 tooth gear because in traveling once around the sun gear each of the planetaries is driven through one revolution about its axis in addition to its motion of translation, and therefore the movement imparted to the ring gear, speaking in terms of the teeth on the internal gear, is 84 teeth + 28 teeth or 112 teeth for each revolution of the 117 tooth gear. $\frac{112}{84} = \frac{4}{3}$.

Because of this step-up of speed through the differential, assuming that the sun gear is held fixed, the ring gear on its outer 117 tooth diameter must mesh with a 62 tooth gear, not a 42 tooth gear on the second shaft which connects through the Hermeflex coupling to the controlling curtain roller of the leading curtain system. With the gears as shown and above described, the rollers of the following curtain and leading curtain will make the same number of revolutions as the drive mechanism propels them. The arithmetical relationships in the gear system will be discussed further in a separate section of this report.

The curtain opening cam and its associated cam follower and gear train are shown in the middle view of Fig. VI. This cam consists of two portions each of which approximates a constant radius connected by a transitional drop and rise.

The cam follower which is a steel roller 3/8 inch diameter is carried between two side plates which form a lever arm pivoted at one end on a shorter lever arm which is held on a transverse shaft passing between the two side plates of the mechanism. At the upper end of this lever arm it is connected to a link which passes across the top of the cam to a gear sector which is part of a 250 tooth, 64 pitch gear of $\frac{1}{4}$ inch face. This gear sector has its bearing in a sleeve mounted around one end of the shaft which provides the bearing for the 175 tooth sector that carries the cam follower for the other cam.

As this cam is rotated, it moves the 250 tooth sector back and forth and this motion is transmitted through three step-up systems, each consisting of a 48 tooth gear and a 96 tooth gear to finally actuate a 37 tooth gear which is on the shaft of the 28 tooth sun gear of the differential. Motion of the 250 tooth sector gear and the other gears of this train in the directions indicated in the drawing causes the ring gear of the differential to move forward in the same direction in which it is propelled by the drive from the main drive cam when it is on the rising part of its curve. Thus an increase in the diameter of the part of the curtain opening cam which is under the cam follower causes the leading curtain to move ahead of the following curtain, thus creating an opening between the two curtains. Conversely when the cam follower is on the smallest diameter of the curtain opening cam and when therefore the 250 tooth gear sector is in the position shown by the dotted lines at the bottom of this middle view of Fig. VI, the 37 tooth gear and the sun gear attached to it are in such a position that the opening between the two shutter curtains is eliminated entirely and there is an overlap of approximately .25 inch.

A series of faint circles and dotted lines show the direction in which the lever which carries the bottom bearing of the cam follower lever is rotated by the spring coiled about its shaft if power is removed from the motor which holds it in the position in which it is shown by the solid lines of the drawing. This auxiliary motor and the gear train by which it drives the shaft of the short lever just referred to are shown in Fig. VI.

The position assumed by the cam follower lever associated with the curtain opening cam when power is removed from the auxiliary motor is shown in dotted lines. The path of the highest point of the cam is shown by a light line indicated by an arrow and a note on the drawing. It may be seen that the cam follower is out of contact with the cam throughout the entire rotation of this cam. The transmitted action of the springs in the curtain rollers will drive the 250 tooth gear to the right until it reaches the position shown by the dotted lines, resting against one of the cross support studs of the frame as a limit stop, and this brings the gear train and the sun gear of the differential to the positions in which there is no curtain opening. As long as the curtain opening cam follower is displaced in this manner, there will be no curtain opening at any point in the cycle, but the two curtains will be driven back and forth in a constant overlapping relationship by the action of the main drive cam and its associated parts. Application of power to the auxiliary motor will return the cam follower lever to its operating position and the shutter will then again open on the forward motion of its cycle and make the exposures predetermined by the cam shapes shown.

Two large scale drawings have been furnished showing the shapes of the two cams accompanied by tables of polar coordinates by which they can be made in the machine shop.

Two large graph sheets show the laws of motion of the two curtains as they move when making exposures. The curve marked "follower curtain" is the curve from which the shape of the main drive cam has been derived; the curve marked "curtain opening" shows the distance between the edges of the leading and following curtains at each stage in the cycle and is the curve from which the shape of the curtain opening cam has been derived.

When the curtain opening cam is rendered inactive by shifting the fulcrum point of its cam follower, the leading and following curtains both follow the law of motion shown for the following curtain in this graph except that an overlap of .25 inch is maintained throughout the cycle.

In these two graph sheets, the scale of degrees runs from 0 to 90 degrees because these curves are plotted in terms of the motion imparted to the synchro resolver which, in accordance with the specification supplied by Perkin-Elmer, is coupled to the main cam shaft at a gear reduction of four to one. Thus the main cam shaft rotates 360 degrees for each 90 degree revolution of the resolver. This gear reduction of four to one is accomplished by the 112 tooth, 18 tooth, and 90 tooth gears shown in the bottom view of Fig. VI. The 90 tooth gear is indicated as the "synchro resolver gear".

The lateral relations of all these gears and cams as they are spaced between the two side plates of the structure are shown in the top view of Fig. VI. The gears of the main drive from the servo motor to the main 140 tooth drive gear together with the main cam shaft and the shaft carrying the two sector gears are supported between ball bearings mounted opposite each other in the two side plates. The gears of the step-up trains transmitting the motion from the cam followers to the finally driven shafts are supported on studs riveted into the side plates. The gears have rather large hubs in which ball bearings are inserted to revolve around these studs. The ends of the studs are chanelled to receive spring washers such as the Waldes "Truarc" washers to hold these gears and bearings in place on the studs.

III. <u>Numerical Relations in Curtain Drive and Gear Trains</u>

a. <u>Curtain Drive</u>.

Rollers - average effective diameter .748; this is 2.350 circumference.

Total curtain travel 23.5 inches. 10 revolutions of rollers required.

Curtain thickness .0013. 10 layers gives .013 radius build-up.

Roller therefore should be .735 diameter and with curtain completely rolled on it will reach .761 diameter.

The movement is 1.74% slow at the small diameter and an equal amount fast at the end. The accumulated error of position is .87% at each end of the run relative to the center position. Compensation for this has been included in the cam figures as given.

b. Gear Train.

The main drive cam turns the gear sector which carries the cam follower 28.8 degrees. If this sector is part of a 175 tooth gear, it will be turned through 14.0 teeth.

Curtain roller drive shaft gear has 42 teeth. This must be turned 10 revolutions or 420 teeth which is 30 x 14 teeth.

Therefore step-up gears between sector and curtain roller drive shaft gear must have an overall ratio of 30.

2.34 which is identically equal to $\frac{117}{50}$ is very nearly the fourth root of 30. $\left(\frac{117}{50}\right)^4 = 29.98$. Therefore the step-up gear train consists of 4 compound gears consisting of a 50 tooth gear and a 117 tooth gear.

C. Drive to Differential.

The opening and closing movement of the leading curtain relative to the following curtain is 6.51 inches or 2.617 revolutions of the curtain roller. The gear which drives the roller has 56 teeth, not 42, because the driving motion is stepped up in the ratio of 4 to 3 by the differential. Therefore 2.617 x 56 = 146.55 teeth to be driven by the 117 tooth external gear of the differential ring gear.

Therefore the motion imparted to the sun gear must cause 146.55 divided by 117 or 1.2526 revolution of the ring gear. This is 105.2 teeth on the internal gear of the ring. The sun gear must rotate the same number of teeth which is 105.2 divided by 28 or 3.757 revolutions of the sun gear.

The shutter opening cam as at present designed imparts 25.00 degrees motion to the gear to which it is linked. Because of the linkage, this sector gear must have a pitch diameter of not less than 3-3/4 inches, preferably more. A suitable pitch diameter is 3.907 inches corresponding to 250 teeth. On this gear 25.0 degrees rotation equals 17.375 teeth.

If a 37 tooth gear is used on the projecting end of the shaft of the sun gear, a movement of 3.757 revolutions will equal 139.01 teeth to be driven.

139.01 divided by 17.375 equals 8.00. Therefore any train of 3 compound gears, each of which gives a step-up of 2 to 1, may be used. The gears shown in the drawing are combinations of 48 and 96 tooth gears, which permit an arrangement in which the two sector gears driven by the two cam followers are mounted on the same axis.

IV. Power and Load Considerations

It is considered essential that the curtains should never touch the glass plates between which they operate. They may be subjected to an accelerational load of 4G, which may be in a direction additive to gravity, so we have computed the required curtain tension in terms of a force of 5G. The problem is the well known one of the force required at the ends of a wire to maintain it with a given amount of sag. (Sokolnikoff - "Higher Mathematics for Engineers and Physicists", page 222.) Taking into consideration the mass of the thin curtain, the 5G force, and allowing a sag of .250 inch in 23 inches, we find the required tension to be 5.98 pounds, or for practical purposes 6 lb.

This means that the springs in the rollers which are connected to the actuating mechanism through the Hermeflex couplings must be wound to a tension of about 5.5 pounds when the curtains are wound up on them. The springs specified below have been designed to give an increase in tension of 0.25 pounds when they are wound 10 turns. Therefore when the curtains are wound all the way in the other direction, these springs will be exerting a tension of 5.75 pounds. The opposing springs must at all times have an excess of tension of at least 0.500 pounds in order to overcome the friction in the Hermeflex and the friction of the curtains in traveling around the cooling tubes shown in Fig. V. If the springs in the purely spring driven rollers are wound to a tension of 6.25 pounds when the curtains are wound up on them, they will exert a tension of 6.50 pounds when the curtains are drawn all the way away from them.

The tension to be overcome by the driving mechanism, then, varies between 0.5 pound and 1.0 pound. To this must be added the frictional torque of the Hermeflex, which is approximately 0.2 pound referred to a radius of .374 inches.

This tension, which is on a radius of .374 on the roller, appears at the teeth of the 42 tooth gear, which has a pitch diameter of .328 inches. The force at the pitch line of this gear, therefore, is $\frac{.374}{.328}$ times 1.2 pounds or 1.37 pounds. Each system of rollers and curtains introduces this maximum force. The gear ratio being 30, the force on the cam follower of the main driving cam, and on the teeth of the sector gear on which it is mounted, reaches a maximum of 82 pounds.

The sector gears and the gears with which they will mate are to be made of hardened tool steel. Calculation on the basis of the generally accepted Lewis formula indicates that a 64 pitch gear of ¼ inch face should sustain this load safely. However, to increase the margin of safety, if a prototype is to be built we will change these gears to 32 diametral pitch and increase the face width to 5/16 inch.

The loading of the other gears in the train with one exception is so much less that they present no problems of strength.

The exception is the mesh between the teeth of the first 117 tooth gear and the 50 tooth gear following it. At this point the load will reach a maximum of 50/117 times 82 pounds or 35 pounds. The faces of these gears should be increased from 1/8 inch as shown in the drawing to 3/16 inch, and these gears also should be made of hardened tool steel. The other gears can be made of mild steel or one of the 400 series stainless steels.

The gears in the train from the shutter opening cam to the 37 tooth gear on the sun gear shaft are subjected to much lower pressures because the total gear ratio in this case is only 8 to 1 instead of 30 to 1. Therefore no special precautions need be taken to avoid failure.

V. Power Requirement

The main drive cam moves its associated sector gear a distance of 14 teeth against a maximum force of 82 pounds in ½ revolution, or 70 teeth of the 140 tooth gear. The maximum rate of rise, however, is twice this average rate, so that the maximum load on the teeth of the 140 tooth gear is not 1/5 of 82 pounds but approximately 2/5, or 33 pounds. This is reduced by a gear ratio of 240 and therefore appears at the teeth of the motor pinion as a load of .138 pounds. Since the radius of the pitch circle of this gear is .078 inch, the load in inch ounces is only .138 x 16 x .078 or 0.172 inch ounce. The motor which has been proposed for this application has a torque rating of 1.5 inch ounces at maximum power output and a stall torque of 2.8 inch ounces. It is clearly more than adequate. However, this analysis ignores friction in the system, and it may not be unreasonable, therefore, to use a motor of this size.

VI. Weight

We have calculated the weight of the construction as closely as possible assuming that all parts are made of steel and assuming a ribbed structure for the side plates and as much reduction of the gear weights as is practical by reducing the thicknesses of the webs back of the teeth and drilling holes to lighten the gears. The result is a weight of 6.78 pounds for the actuating mechanism including the servo motor, the synchro resolver, and the auxiliary motor. The weight of the curtain rollers, the curtains, the support brackets for the curtains, and the hermeflex couplings will be close to 4.65 pounds, making the total weight of the curtain system, apart from the frame in and on which it is to be installed, 11.43 pounds.

If the side plates, spacing posts, and as many of the shafts as possible are made of titanium it appears that about $2\frac{1}{4}$ pounds can be saved. The saving by the use of titanium is not impressive, since it is probably not practical to make the gears or the curtain rollers of this material.

VII. Curtain Spring Specification

On the basis of formulas given in Wohl, "Mechanical Springs" we have found that the following springs will probably do about what is required in the curtain rollers:

.026 diameter piano wire.

No. of turns, free -- 142.

Maximum winding, 260 turns.

Diameter of helix, free, 1.009 inches.

Diameter, fully wound .356 inch.

Tension, fully wound, measured on circumference of roller, 6.5 pounds.

Change in tension with 10 turns, 0.25 pounds.

Obviously the spring will have to be partly wound to get it inside the hollow roller, but this is of no consequence as it is always used nearly fully wound. Fully wound it will have a little play around the .312 diameter center spindle of the curtain roller.

The maximum stress in the spring wire is slightly less than the maximum value recommended for this type of wire by Wohl.

VIII. Reliability

The least reliable elements in the system are probably the Hermeflex couplings. The engineer in charge of design of these units recommended that they be changed after each 5000 cycles of operation. Failure would probably be due to lubrication failure, since the construction does not lend itself to good lubrication of the two bearings at the ends of the unit. Furthermore, the inner bearing must be lubricated with Dow Corning High Vacuum grease, which is not an outstandingly good lubricant.

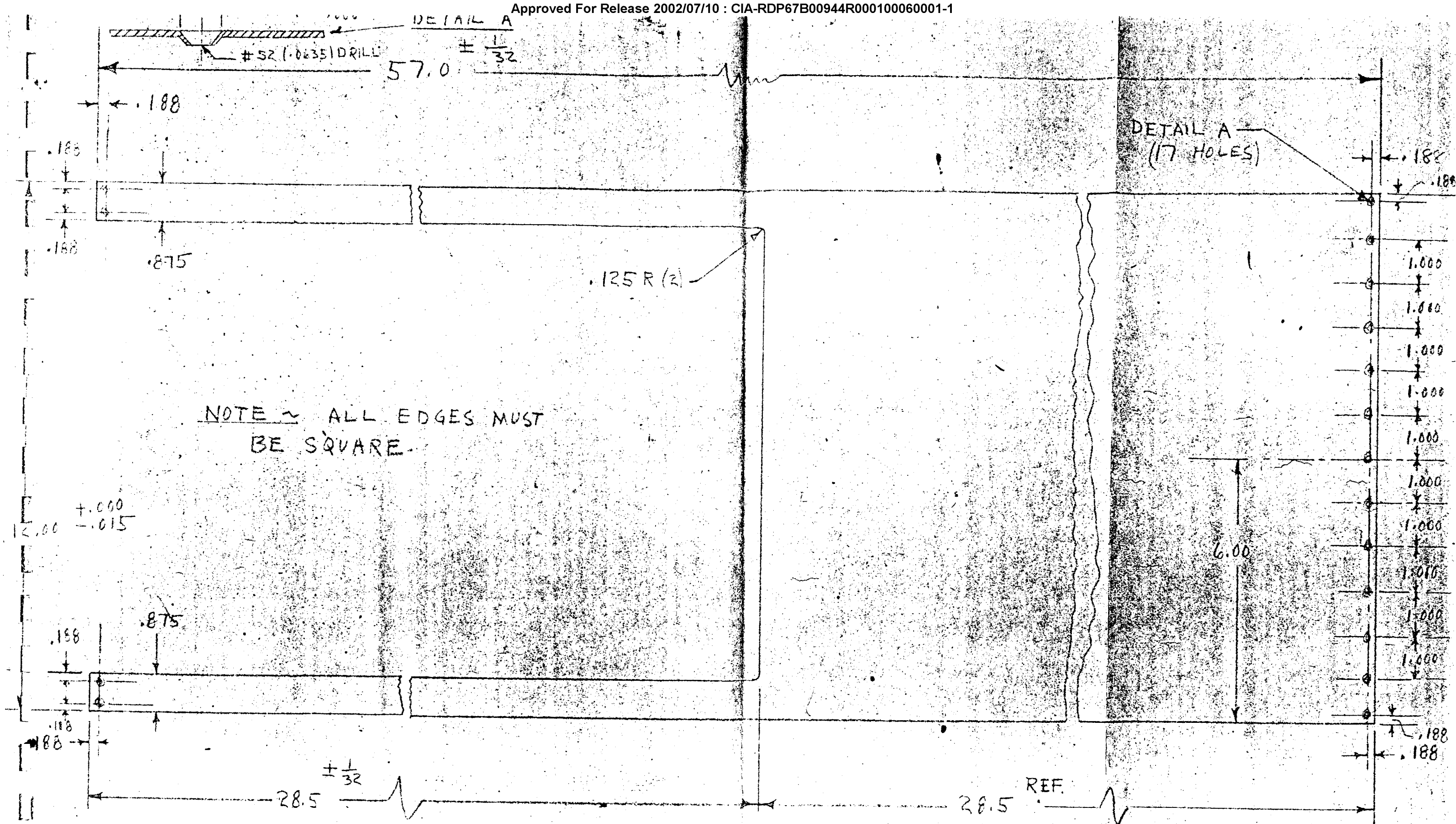
The curtains are unlikely to fail, since the tensions applied to them are only a small fraction of the tensile strength of the ribbons. The curtains move slowly and are never subjected to abrupt changes of velocity, therefore they will not be subjected to any disruptive stresses. Similar curtains used in other structures in the past have shown that they will not fail due to flexure alone, except possibly after several hundred thousand cycles of operation.

The ball bearings in the system are not subjected to heavy loads with three exceptions. These are the bearings on the "main drive cam" side of the structure on the main drive cam shaft, on the cam follower shaft, and on the shaft of the first gear in mesh with the cam follower. These bearings are loaded almost to the limit indicated in the Barden catalog as the acceptable loading for 500 hours design life at low speeds. 500 hours life at 100 rpm would be 3 million revolutions. Since the stated "average life" of the bearings under these conditions is 2500 hours it does not appear that these bearings are likely to fail. However, the effect of the moderately high temperature environment cannot be predicted, since no data are available.

The cams, cam followers, and gears may be expected to last for very long times if they are made of good tool steel properly heat treated.

The expected life of the servo components is not indicated by the manufacturer.

STATINTL



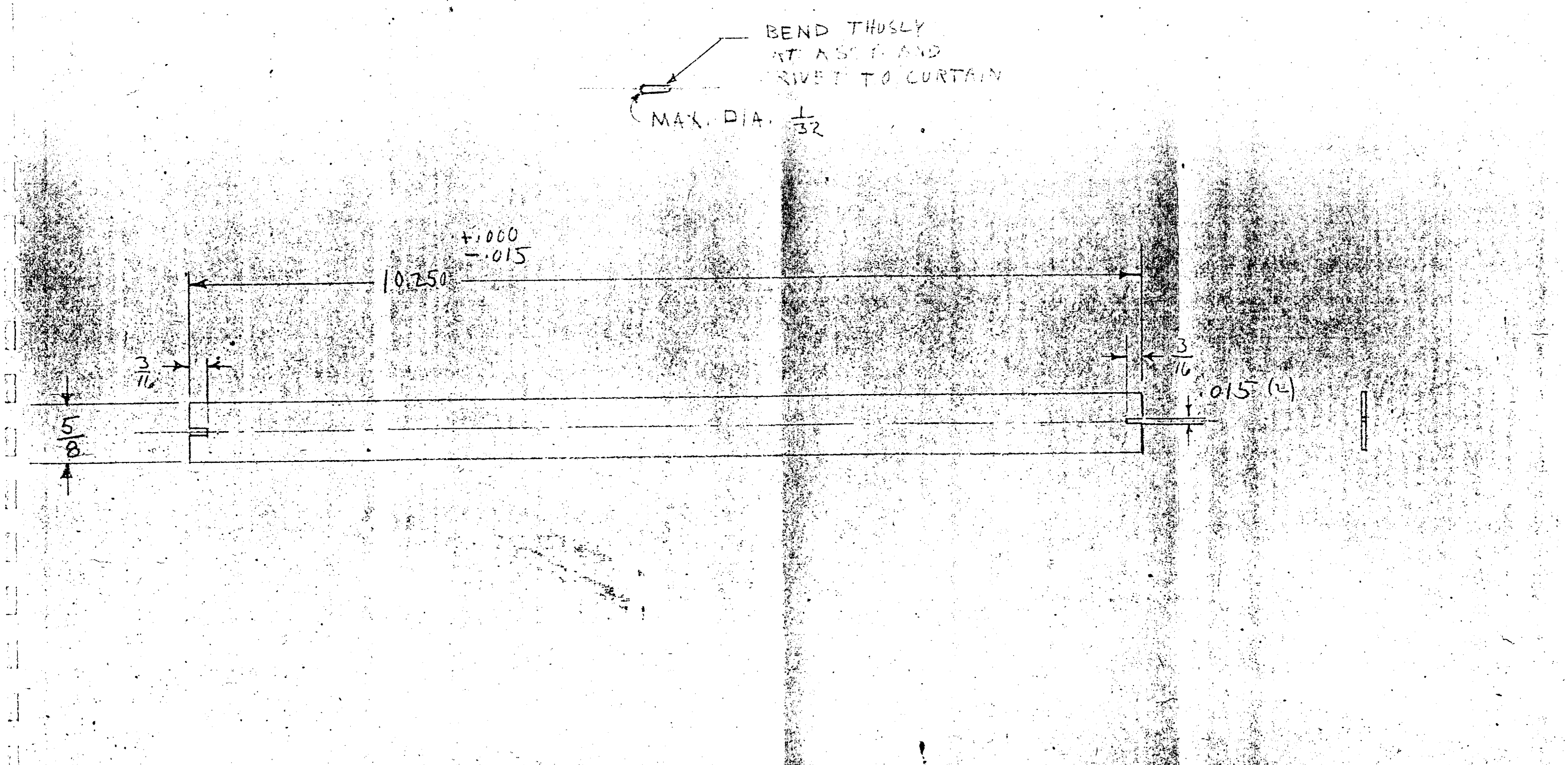
NOTE ~ MATERIAL HARD ROLLED STAINLESS STEEL TYPE 302 .0013 ± .0001 THICK

SCALE ~ HALF SIZE

FIG. I

2 REQ'D.

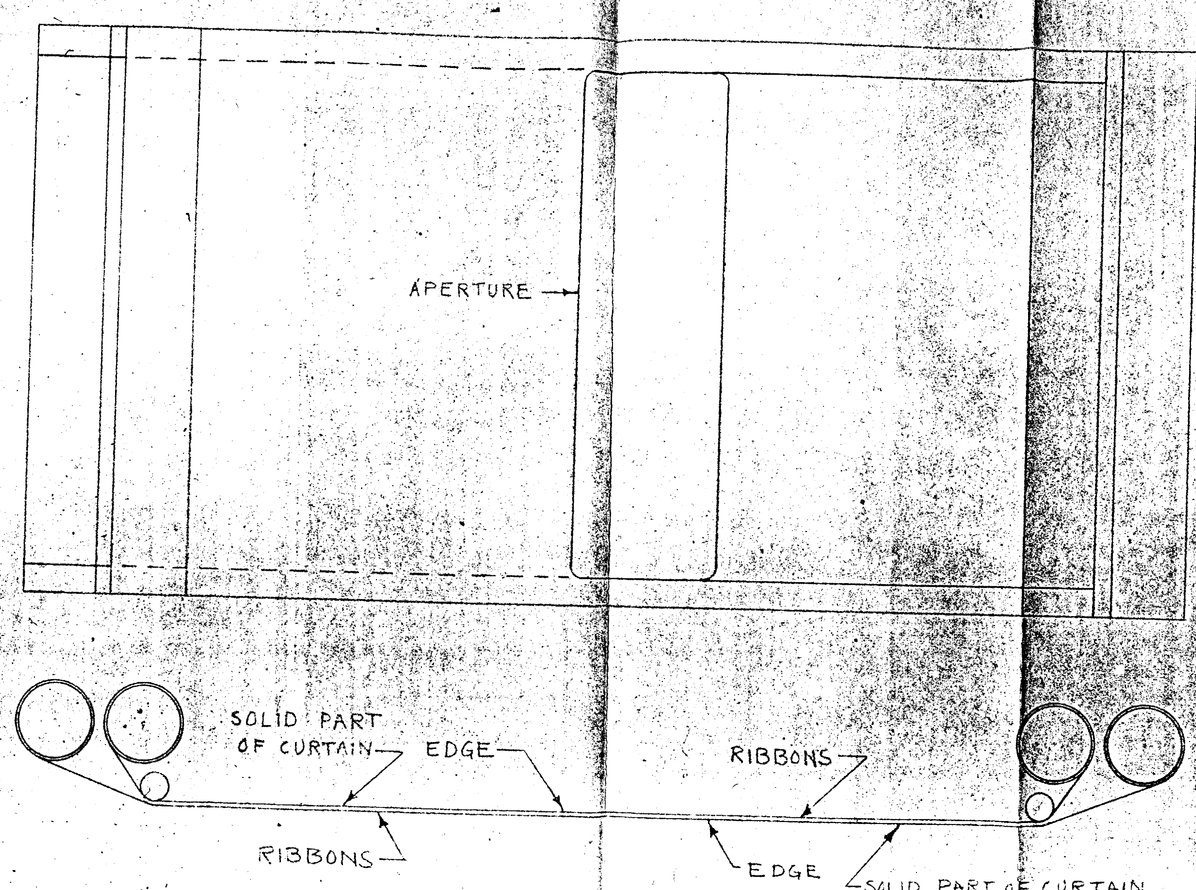
CURTAIN



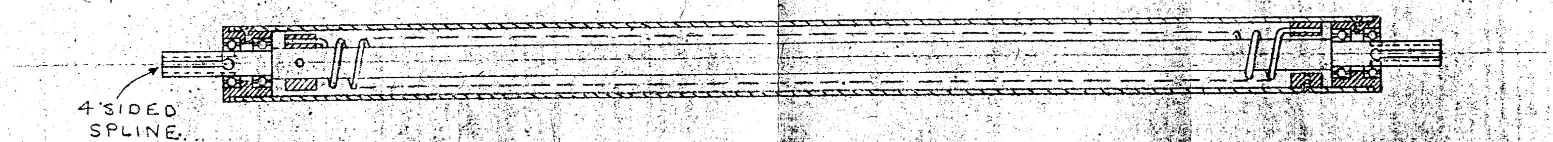
MATERIAL ~ STAINLESS ST. 302 .003 THICK 2 REQ'D. SCALE ~ FULL SIZE

FIG. II

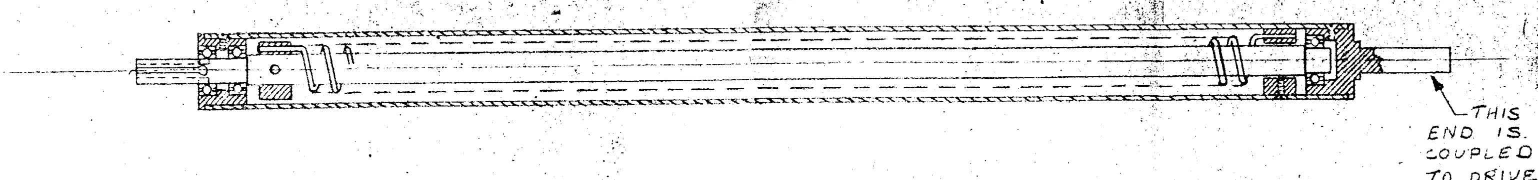
CURTAIN EDGE REENFORCEMENT



SCHEMATIC DRAWING OF CURTAIN AND ROLLER CONSTRUCTION FIG. III



CURTAIN ROLLER — SPRING DRIVEN. (ON END OPPOSITE DRIVE MECH.)

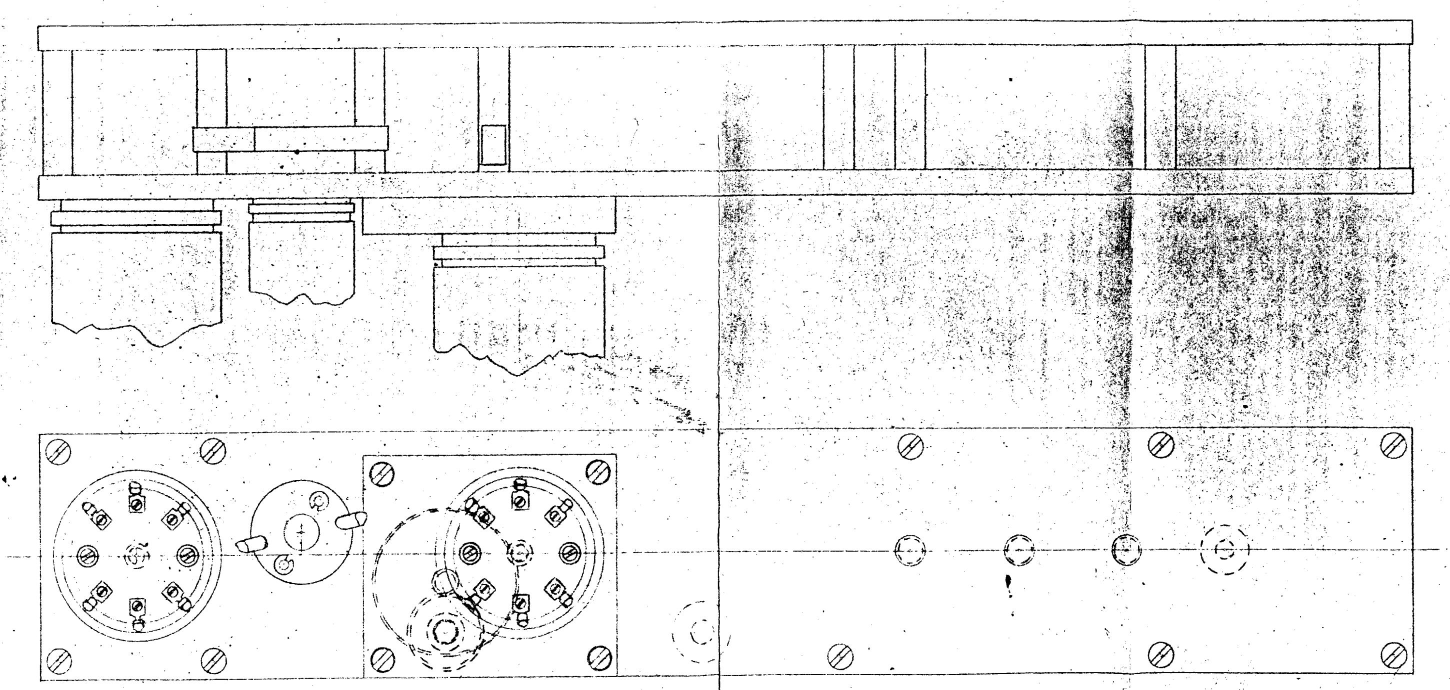


CURTAIN ROLLER — POWER DRIVEN, WITH SPRING TO MAINTAIN TENSION.

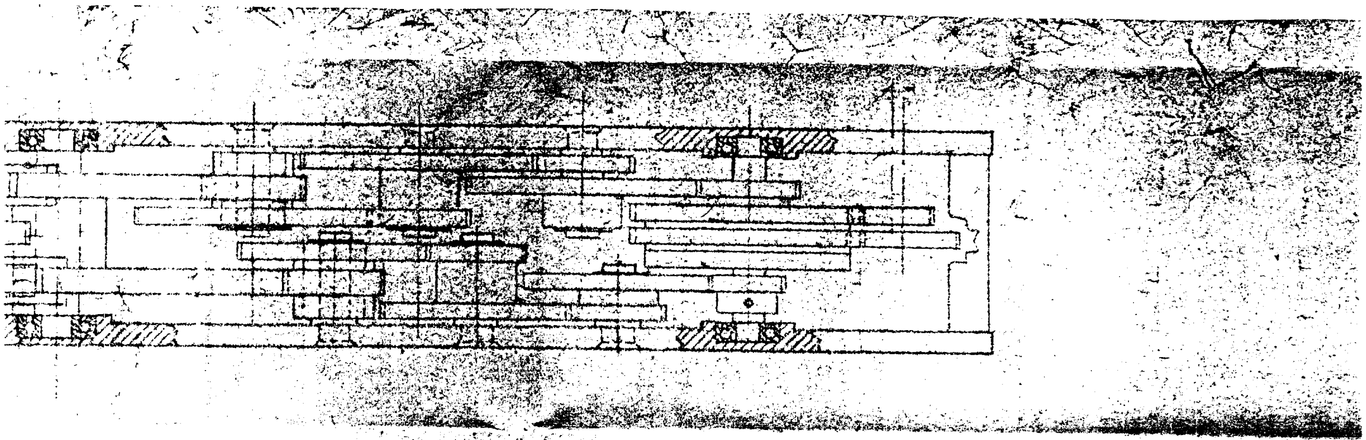
THESE SPRINGS ARE NECESSARILY EXPERIMENTAL. AS A STARTING POINT WIND #18 (.041 DIA.) MUSIC WIRE TO GET A COIL 10" LONG WITH 16 TURNS PER INCH AND I.D. .437" ± .016.

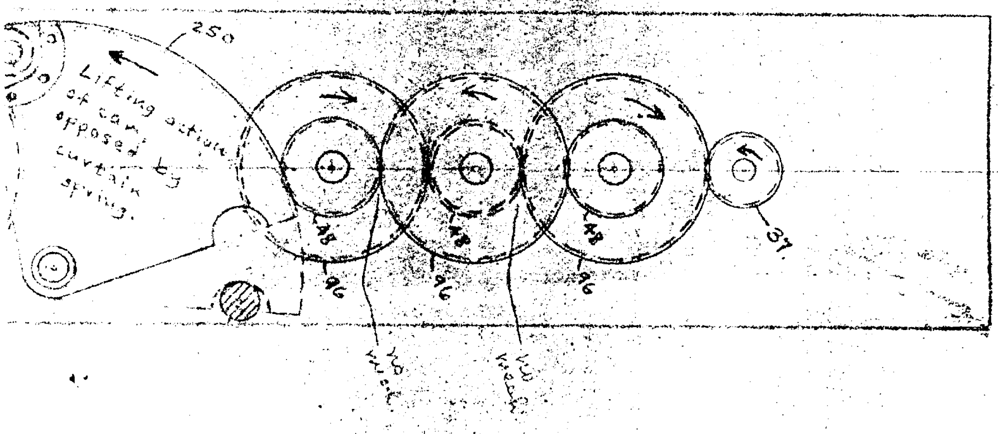
FIG. IV

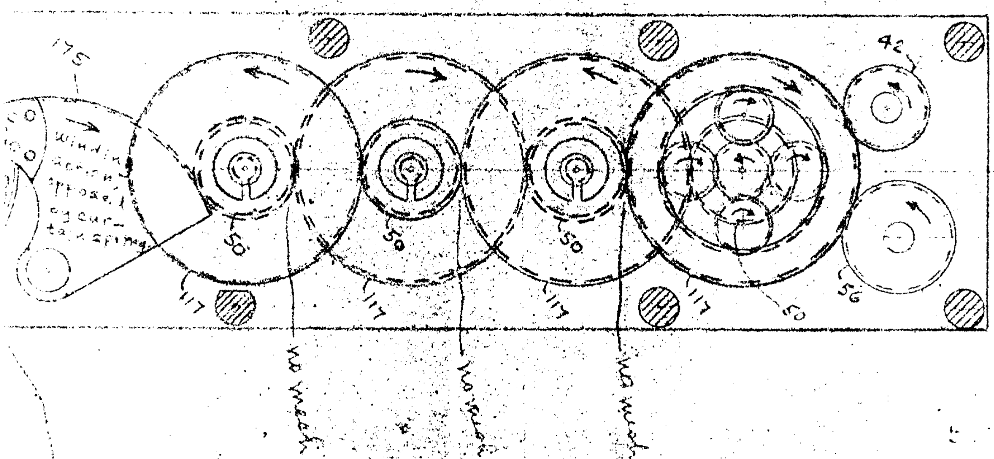
CURTAIN ROLLERS



OUTSIDE VIEWS OF MOTORS AND FRAME, WITHOUT GEARS ON INSIDE.







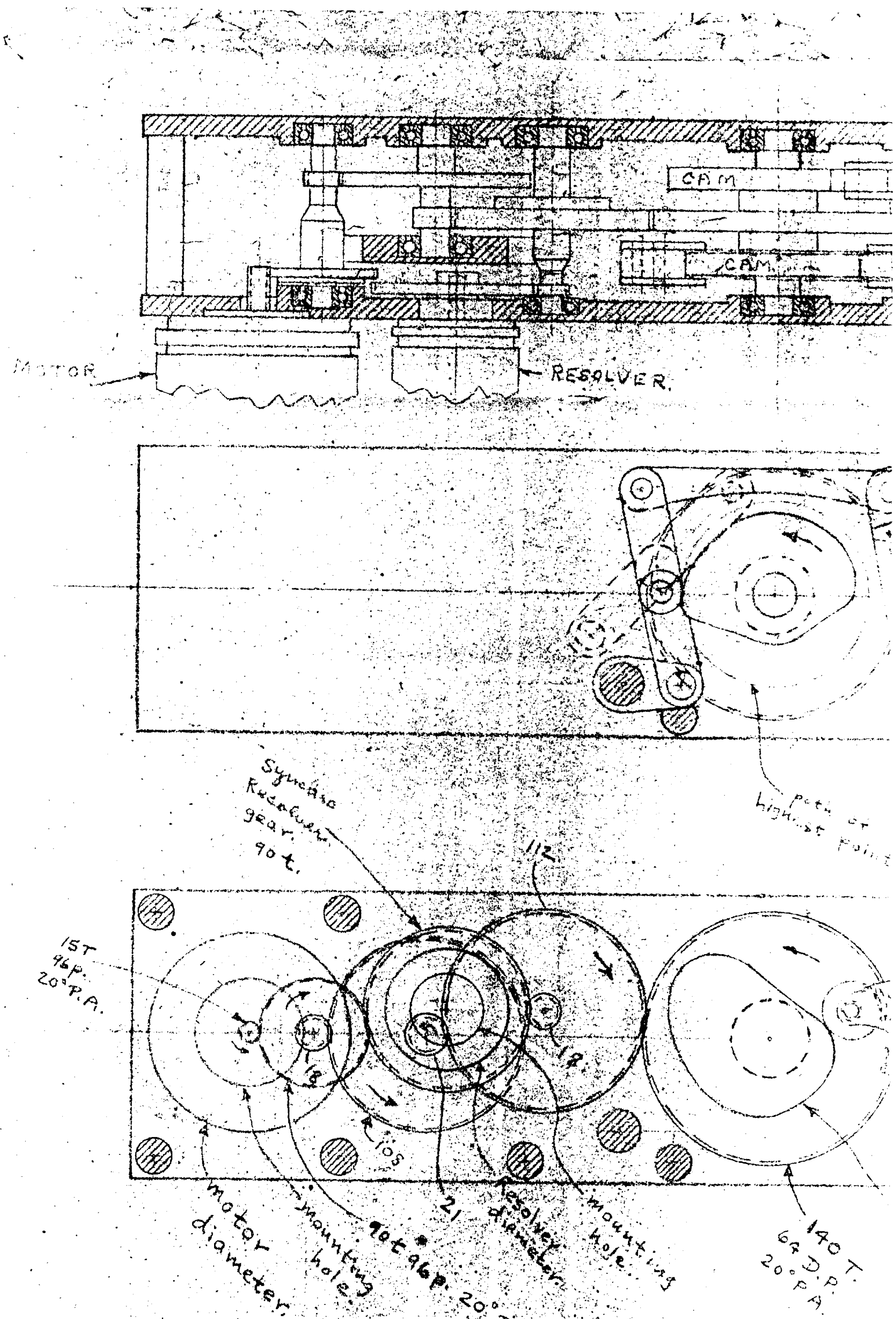
This cam and the follower really ought to be
n in dotted lines, since they are behind the gear
his view.

CAMS AND GEAR TRAINS WITH SUPPORTING PLATES.

IN AND MAIN SHUTTER DRIVE CAM AND GEAR TRAIN.

M AND GEAR TRAIN TO SUN GEAR OF DIFFERENTIAL.

 FIG. VI

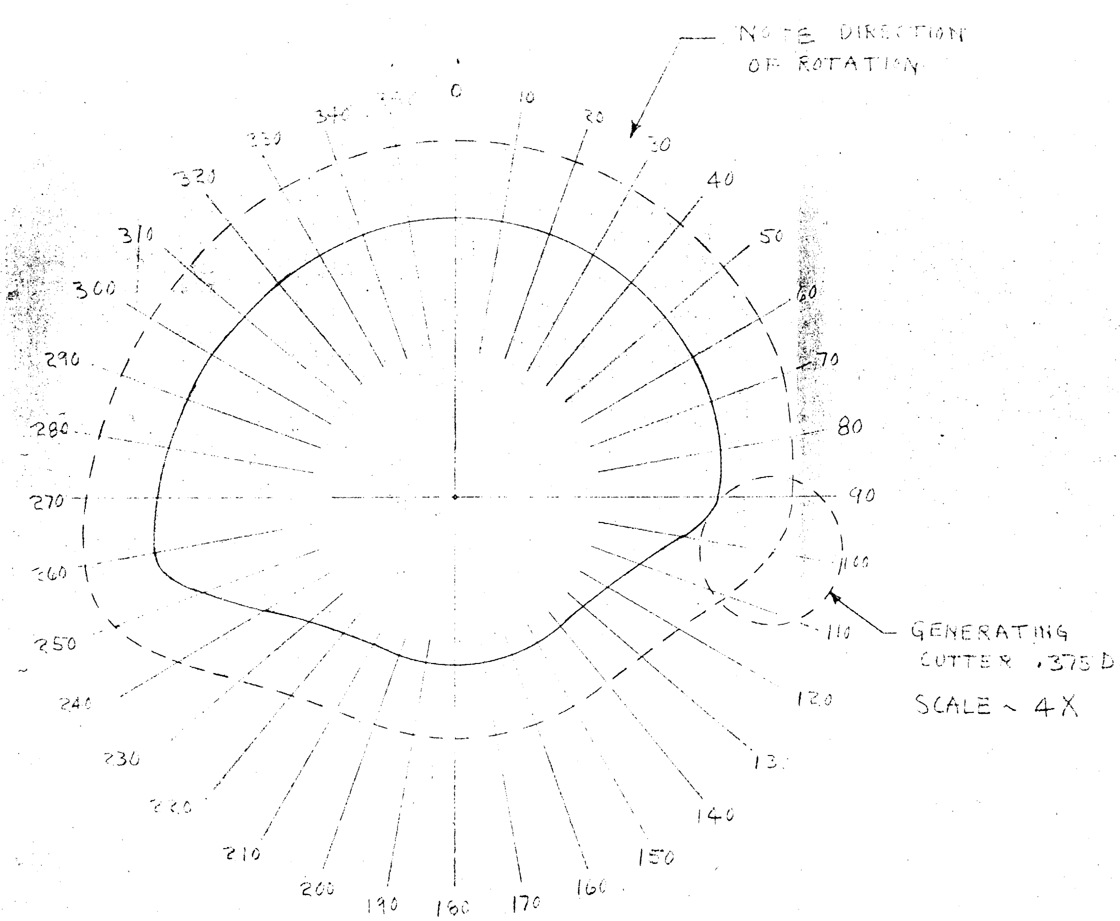


show
in t

TOPMOST VIEW SHOWS EDGEWISE VIEW OF

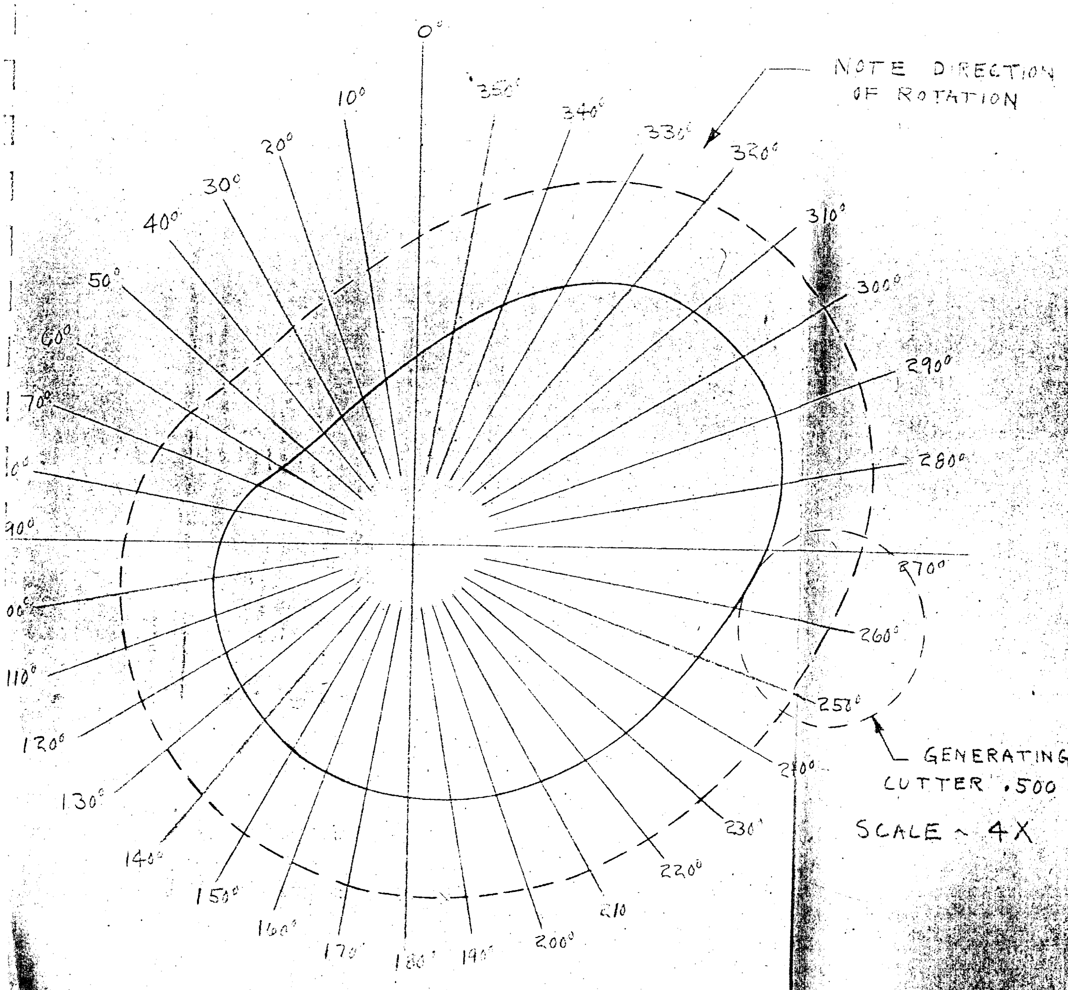
BOTTOM VIEW SHOWS MOTOR DRIVE GEAR TRA

MIDDLE VIEW SHOWS CURTAIN APERTURE CA



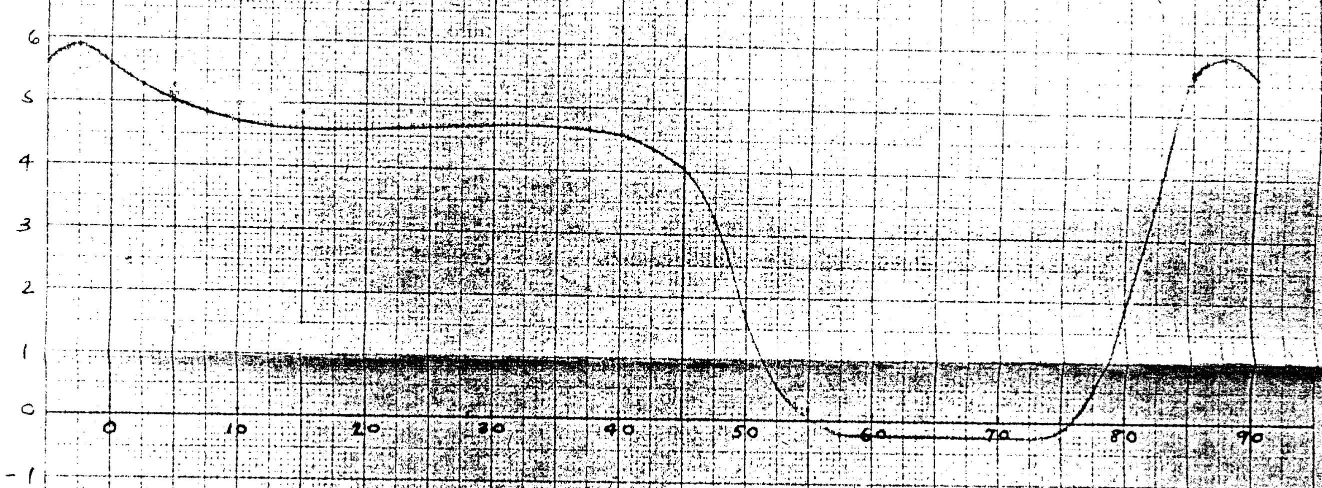
ANGLE θ	RADIUS INCHES	ANGLE θ	RADIUS INCHES
0	.902	180	.612
5	.900	185	.612
10	.902	190	.612
15	.904	195	.612
20	.906	200	.613
25	.906	205	.614
30	.906	210	.621
35	.906	215	.634
40	.907	220	.653
45	.907	225	.685
50	.907	230	.723
55	.905	235	.775
60	.905	240	.830
65	.905	245	.893
70	.902	250	.949
75	.897	255	.976
80	.889	260	.983
85	.883	265	.974
90	.879	270	.965
95	.863	275	.953
100	.833	280	.943
105	.790	285	.934
110	.744	290	.928
115	.704	295	.931
120	.670	300	.918
125	.648	305	.912
130	.631	310	.908
135	.620	315	.908
140	.613	320	.906
145	.613	325	.906
150	.612	330	.905
155	.612	335	.904
160	.612	340	.903
165	.612	345	.904
170	.612	350	.903
175	.612	355	.903

SHUTTER CURTAIN OPENING CAM - PATH OF GENERATING CUTTER AND COORDINATES OF AXIS OF CUTTER



ANGLE θ°	RADIUS INCHES	ANGLE θ	RADIUS INCHES
0	.777	180	.897
5	.740	185	.905
10	.708	190	.913
15	.684	195	.921
20	.661	200	.929
25	.645	205	.938
30	.631	210	.947
35	.621	215	.956
40	.617	220	.967
45	.619	225	.979
50	.628	230	.993
55	.643	235	1.007
60	.664	240	1.022
65	.683	245	1.039
70	.699	250	1.061
75	.715	255	1.087
80	.729	260	1.114
85	.739	265	1.143
90	.749	270	1.174
95	.758	275	1.199
100	.769	280	1.220
105	.780	285	1.231
110	.790	290	1.239
115	.800	295	1.238
120	.808	300	1.231
125	.812	305	1.214
130	.821	310	1.193
135	.830	315	1.175
140	.839	320	1.148
145	.847	325	1.110
150	.854	330	1.066
155	.860	335	1.018
160	.867	340	.965
165	.874	345	.913
170	.883	350	.864
175	.891	355	.821

MAIN CURTAIN DRIVE [illegible]


6
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4
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10
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40
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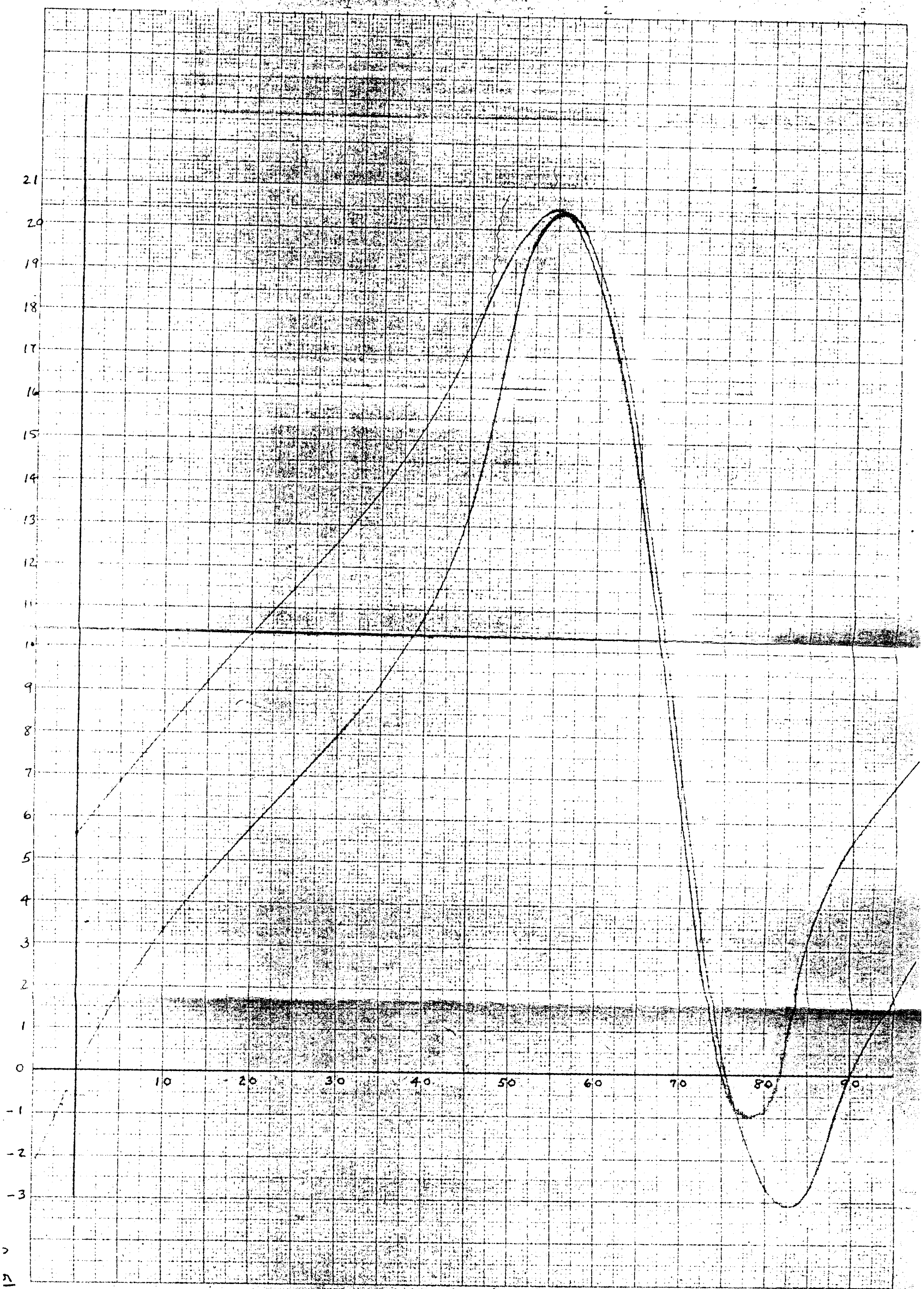
90


21
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-3
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30
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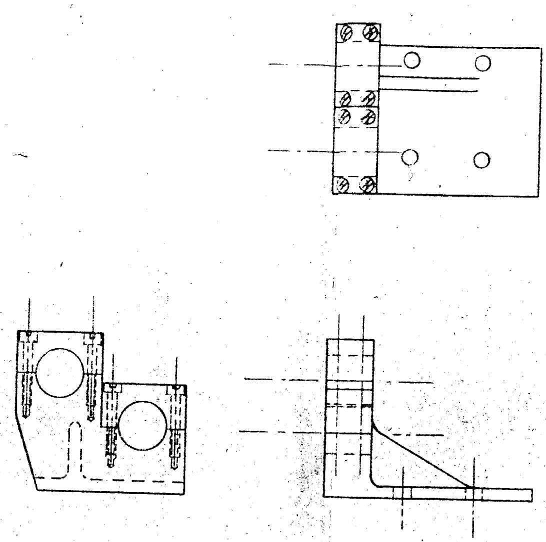
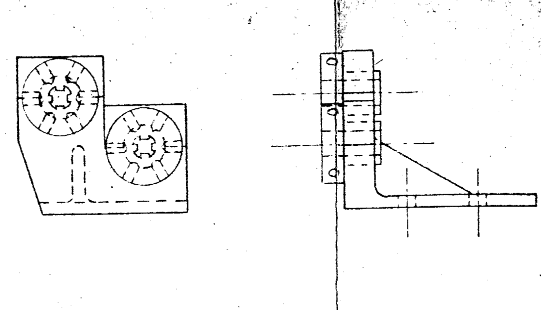
80
90



SUPPORT BRACKET FOR CURTAIN ROLLER — MOUNTS TWO BALL BEARINGS WITH SPACER. ONE REQUIRED.

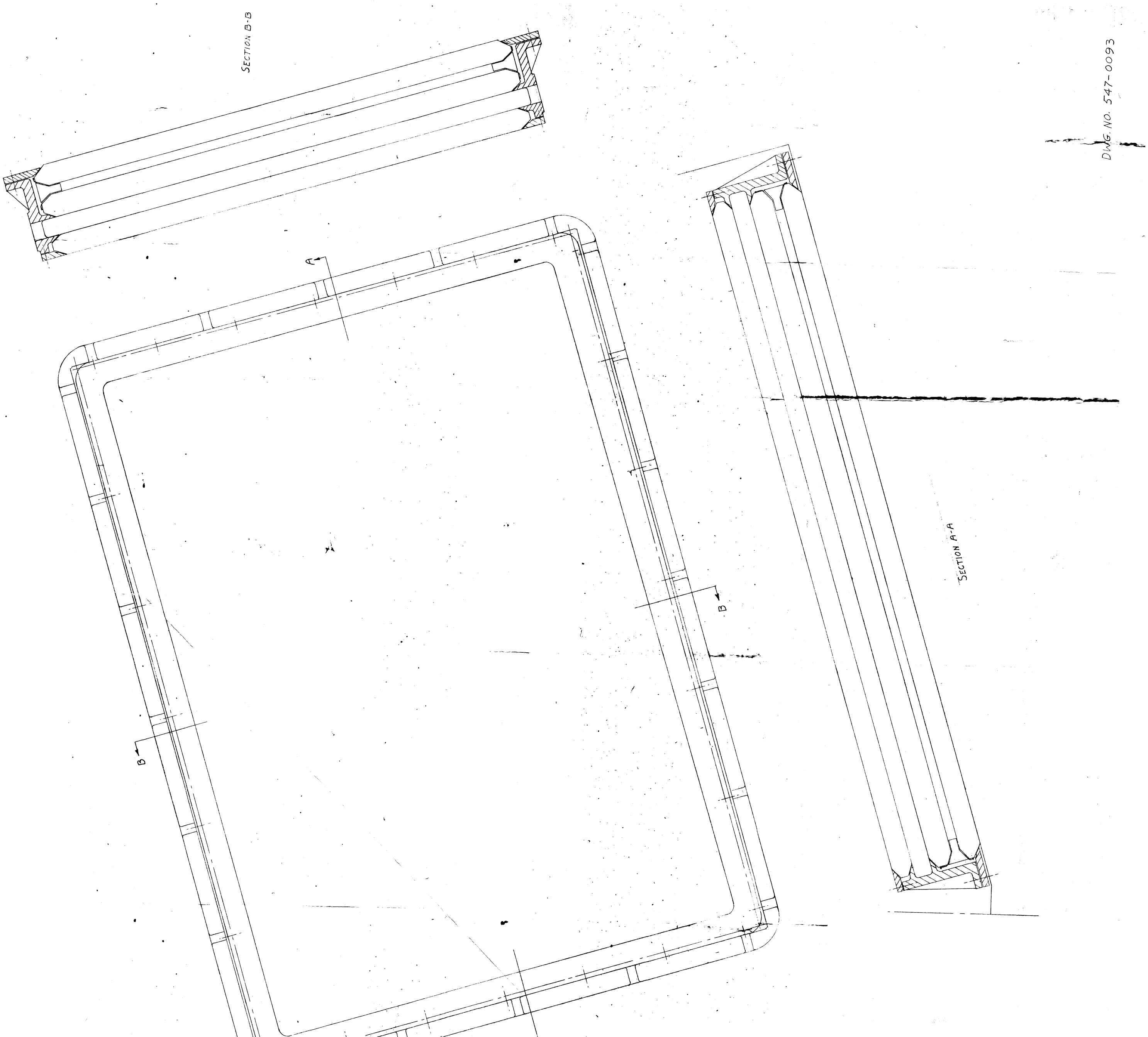


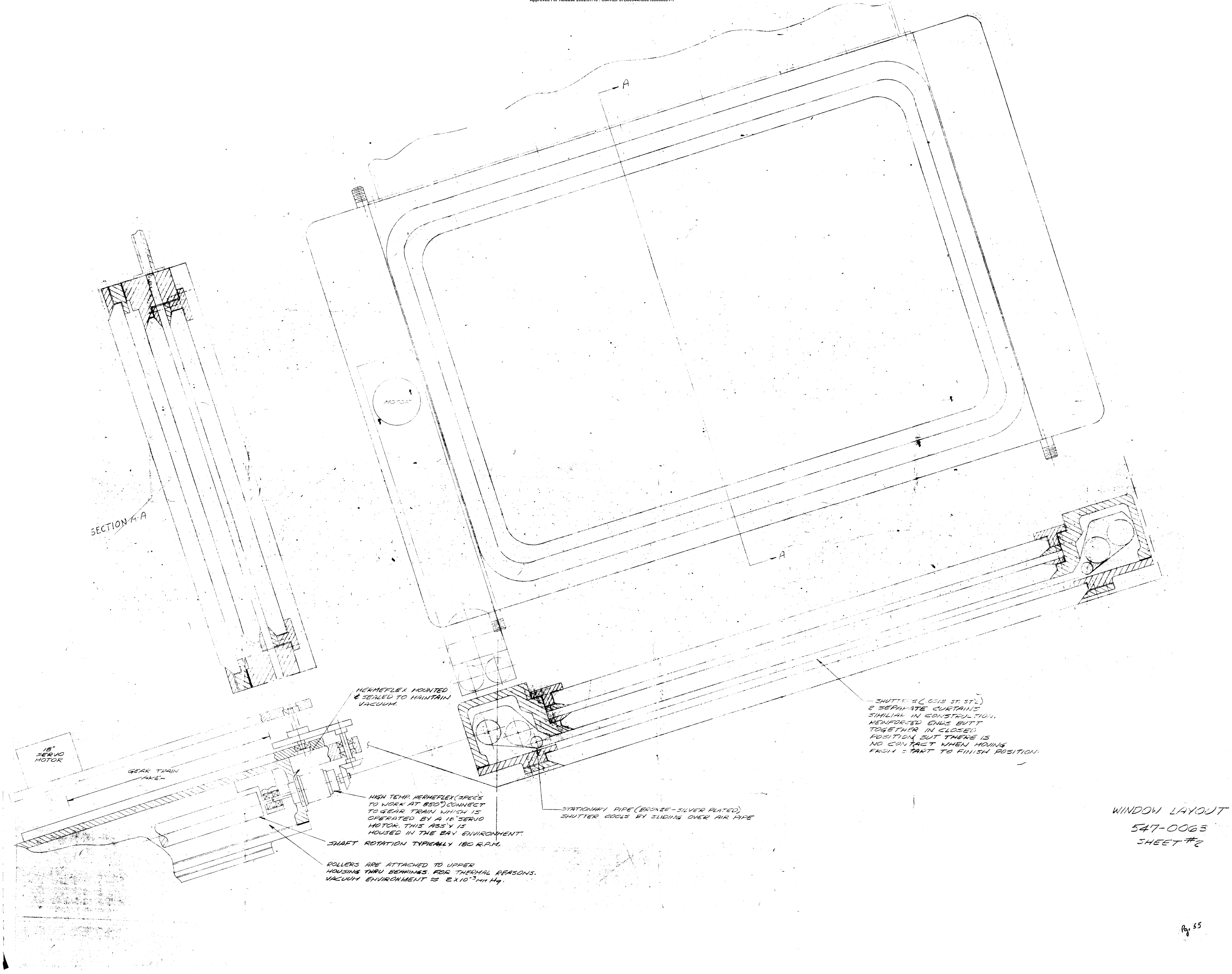
SUPPORT BRACKET FOR CURTAIN ROLLER —
PROVIDES FOR SPRING TENSION ADJUSTMENT.
ONE REQUIRED AS DRAWN AND TWO MIRROR IMAGES.

FIG.

VI PRELIMINARY DESIGN OF WINDOW MOUNT INCORPORATING SHUTTER

This section contains two drawings. Drawings Number 547-0063, sheet 2 is a preliminary window mount design layout incorporating a shutter mechanism. Drawing Number 547-0093 is a preliminary window mount design layout for a configuration which does not incorporate a shutter. It is included as a basis of comparison.


SECTION B-B
SECTION A-A
A
A
B
B
DWG. NO. 547-0093


A
A
MOTOR
SECTION A-A
HERMEFLEX MOUNTED & SEALED TO MAINTAIN VACUUM.
18" SERVO MOTOR
GEAR TRAIN AREA
HIGH TEMP. HERMEFLEX (SPEC'S TO WORK AT 850°) CONNECT TO GEAR TRAIN WHICH IS OPERATED BY A 18" SERVO MOTOR. THIS ASS'Y IS HOUSED IN THE BAY ENVIRONMENT.
SHAFT ROTATION TYPICALLY 180 R.P.M.
ROLLERS ARE ATTACHED TO UPPER HOUSING THRU BEARINGS FOR THERMAL REASONS. VACUUM ENVIRONMENT IS 2 X 10^-5 MM Hg.
SHUTTERS (.0013 ST. ST'L) 2 SEPARATE CURTAINS SIMILIAR IN CONSTRUCTION. REINFORCED ENDS BUTT TOGETHER IN CLOSED POSITION BUT THERE IS NO CONTACT WHEN MOVING FROM START TO FINISH POSITION.
STATIONARY PIPE (BRONZE-SILVER PLATED) SHUTTER COOLS BY SLIDING OVER AIR PIPE
WINDOW LAYOUT
547-0063
SHEET #2

Pg. 55

VII CONCLUSIONS AND RECOMMENDATIONS

If the implications of incorporating a thermal shutter in the window system are consistent with the overall system concept, consideration should be given to fabrication of a shutter prototype. The following points should be considered in deciding whether to fabricate a prototype of the shutter mechanism:

1. The effect of the increased size of window mount.
2. The consequence of increased weight.
3. The importance of increasing light transmission through the window by reducing losses due to low emissivity coatings.
4. The effect of a lateral thermal gradient upon ultimate system performance.
5. The desirability of reducing heat transfer to inner glazings.

Data required to evaluate points 1 and 2 is contained in Section II of this report. Work is presently proceeding on a laboratory model of a window system which is designed to yield information on the magnitude and shape of lateral thermal gradients which would result from various cooling air flows through the inner gap. However, interpretation of the resultant gradient data in terms of system performance may be extremely difficult. The time required to complete these experiments indicates that a decision should be made on the shutter prototype prior to their completion.

The most valid evaluation of shutter reliability and performance in a hot-vacuum environment can be made on a shutter prototype.

[redacted] has been requested to submit a quotation for building a prototype of the mechanism described in Section V of this report. Their quotation is based on our supplying a vacuum tight box of substantially the dimensions of the vacuum gap and frame which will be used in the final window. They will mount the shutter curtain rollers and curtains in this frame, fasten into the frame the two cooling tubes, and install the Hermeflex couplings properly connected to the curtain rollers. They will construct and mount the drive mechanism in proper relationship with the Hermeflex couplings and curtain mechanism in the frame. They will deliver this mechanism to us in operating condition.

Their proposal has been received.

It is recommended that this report be used as a guide to determine whether or not the thermal shutter should be given further consideration; and if so, whether a prototype should be constructed at this time.

The final design submitted in fulfillment of the design subcontract is contained in Section V of this report.

Photomechanisms, Incorporated

The following is a status report submitted by Photomechanisms at our request. The financial section of their report has been deleted herein.

February 5, 1960

STATINTL

Attention: [redacted] - Purchasing

Subject: Status Report for Curtain Shutter Program

Reference: Your P. O. No. J-56119; Our Job No. 526

Gentlemen:

As per your request we are herewith submitting a status report for the referenced program.

1.0 Technical Report. -

Photomechanisms was proceeding, on an overtime basis, as we had proposed in our letter of January 15, 1960. The general specification prepared by [redacted] on January 18 was also being closely followed.

STATINTL

On January 20 a meeting was held at during which the specification and design approach were carefully reviewed. A new space envelope for the mechanism was presented at the meeting by [redacted]. Also, the additional requirement for no make-break contacts, was voiced by on the basis that contacts could not satisfy their rigid acceleration and radio-noise specifications.

STATINTL

Following the meeting, and after careful engineering

February 5, 1960

STATINTL

Attention: [redacted] Purchasing

Reference: Your P. O. No. J-56119; Our Job No. 526

investigation, Photomechanisms indicated that the proper selection of switches would not jeopardize these specifications. agreed to let [redacted] STATINTL proceed on the basis of this judgment.

Enclosed are copies of two layout drawings and a schematic, representing the beginnings of our approach to the problem as of the stoppage date, January 25, 1960. Considerable engineering had also been spent as shown on the following "expenditures to date" statement.

2.0 Financial Report. -

STATINTL [redacted] respectfully wishes to urge that a technical meeting be held as soon as possible in order to re-establish

February 5, 1960

STATINTL

Attention: [redacted] - Purchasing

Reference: Your P. O. No. J-56119; Our Job No. 526

the engineering approach to the problem. After such a meeting is held we would be in a better position to review the contract price.

Very truly yours,

[redacted] STATINTL

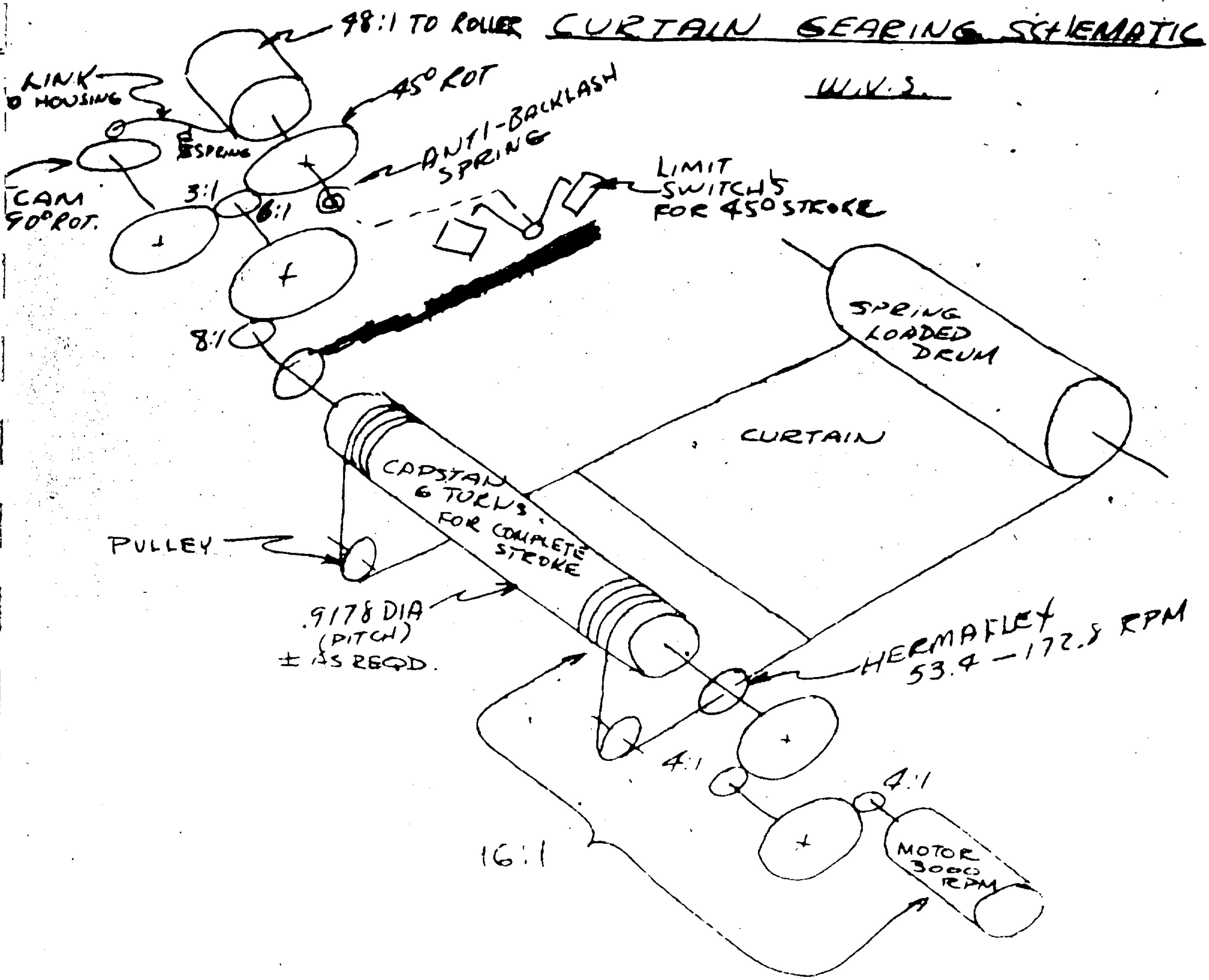
FEJ:mc
Enclosures

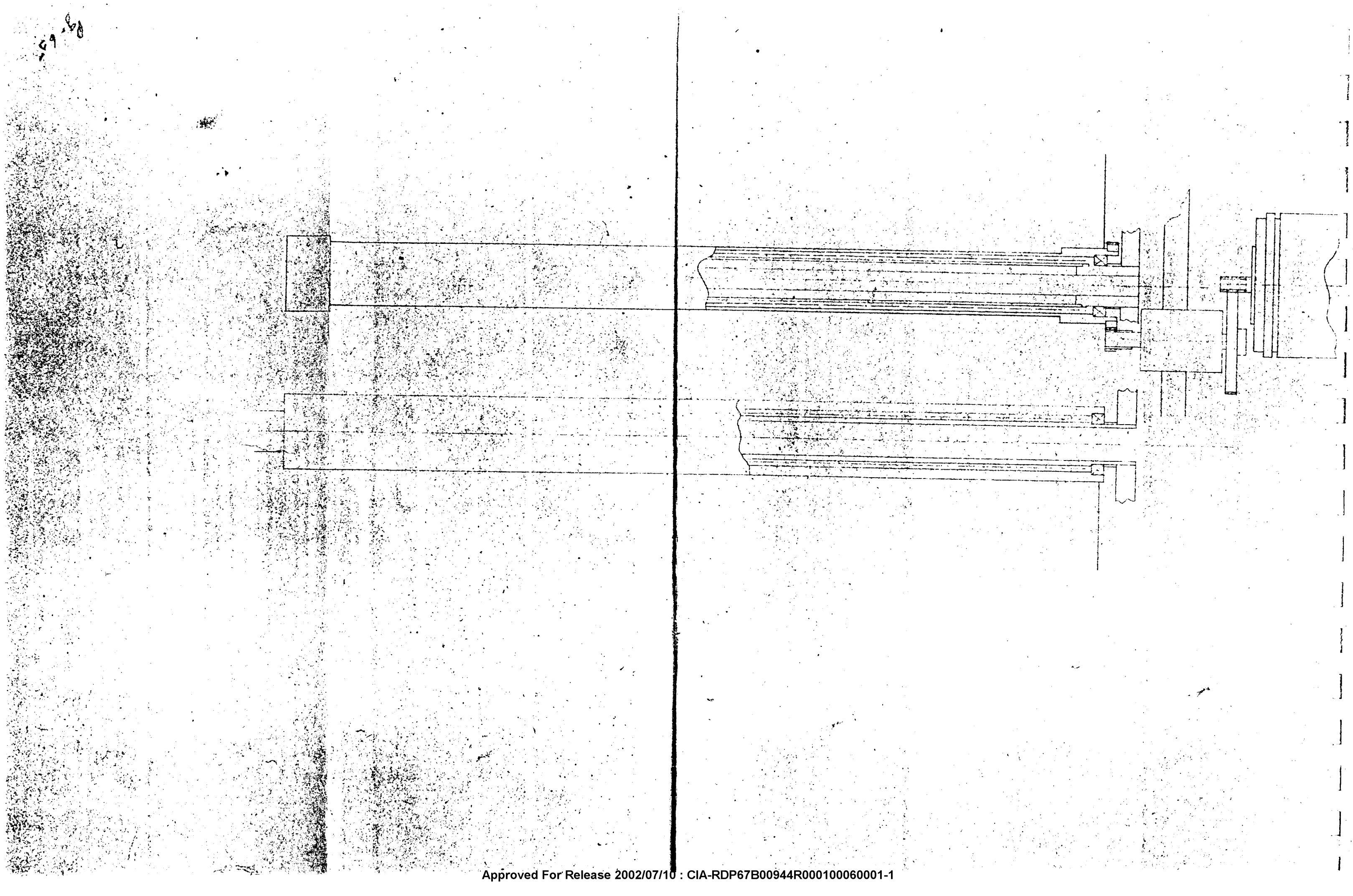
526

STATINTL

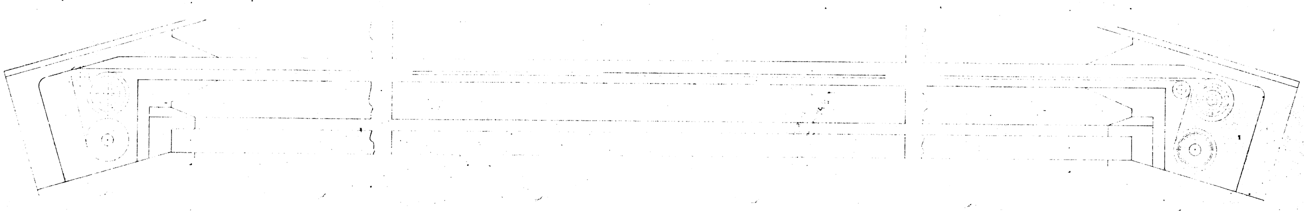
CURTAIN GEARING SCHEMATIC

W.V.S.





Approved For Release 2002/07/10 : CIA-RDP67B00944R000100060001-1



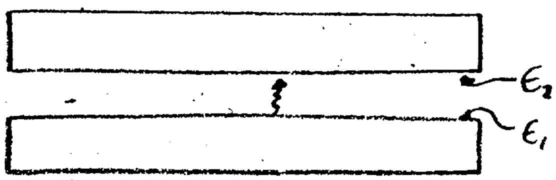
APPENDIX E

EFFECTIVENESS OF SHUTTER TO REDUCE RADIANT HEAT TRANSFER

The thermal configuration employing a shutter can be represented by two discreet cases:

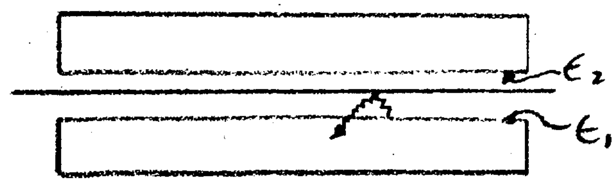
CASE A

Shutter Open



CASE B

Shutter Closed



The general radiation equation is:

$$Q = \sigma A \frac{\epsilon_1 \epsilon_2}{\epsilon_1 + \epsilon_2 - \epsilon_1 \epsilon_2} \left(\theta_1^4 - \theta_2^4 \right) \qquad (1)$$

where:
- σ = Stefan-Boltzman constant
- A = area
- ϵ_1 = emissivity at first surface
- ϵ_2 = emissivity at second surface
- θ_1 = temperature at first window
- θ_2 = temperature at second window

For Case A, the radiant heat transfer is expressed by:

$$Q_A = \sigma A \epsilon_{eff} \left(\theta_1^4 - \theta_2^4 \right) \qquad (2)$$

where $$\epsilon_{eff} = \frac{\epsilon_1 \epsilon_2}{\epsilon_1 + \epsilon_2 - \epsilon_1 \epsilon_2}$$

BEST COPY
Available

For Case B, with the shutter intercepting the radiant energy:

$$Q_B = 0 \qquad (3)$$

The total radiant heat reaching the inner plate is given by:

$$Q = \frac{Q_A t_1 + Q_B t_2}{t_1 + t_2} \qquad (4)$$

where t_1 = fraction of shutter open time, as seen by a point in the inner plate.

t_2 = fraction of shutter closed time, as seen by a point on the inner plate.

Substituting $Q_B = 0$

$$Q = \frac{Q_A}{1 + k} \qquad (5)$$

where $k = \frac{t_2}{t_1}$

Examination of eq.(5) indicates that the relative effectiveness of the shutter can be evaluated by comparing

$$\frac{\epsilon_{eff}}{1 + k} \qquad (6)$$

since $k = 0$ for cases containing no shutter.

For the shutter discussed in Section V,

$$t_1 = .17$$

$$t_2 = .83$$

Therefore $1 + k = 5.9$

Evaluating eq.(6) for various cases, we obtain

Condition \ Surface Emissivity	ϵ_1 .2	ϵ_2 .2	ϵ_1 .2	ϵ_2 .8	ϵ_1 .8	ϵ_2 .8
No Shutter $1 + K = 1$	.11		.19		.67	
Shutter $1 + K = 5.9$	.019		.032		.11	

Approved For Release 2002/07/10 : CIA-RDP67B00944R000100060001-1